QUANTIFICATION AND ANALYSIS METHODS AIMED AT THE REDUCTION OF FLOW INDUCED NOISE IN THE EXPANSION DEVICE

BY

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THESIS

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Abstract

A cultural context where acoustic technologies are rapidly growing, and the demand for comfort is increasing exponentially, were all catalysts for the birth of this research project. In particular, noise is one of the major concerns that many companies want to come against, especially in the HVAC & Refrigeration sector. Indeed, noise coming from air conditioning systems can be perceived as very annoying. Thus, one of the main reasons this research is conducted is to increase consumer comfort in all applications where air conditioning noise, especially noise coming from expansion devices, is perceived as very disturbing. At the same time another motivation to conduct research in this area is to improve the understanding of flow induced noise.

The objective of the thesis is to provide a basis for future scientists who would like to come against the problem of noise produced from expansion devices.

Basic theories regarding noise and expansion devices are defined and explained. Subsequently the thesis demonstrates how the laboratory was designed and setup (the refrigerant used was R134a), including pressure, acceleration, and microphone measurements. The importance of the anechoic chamber will be shown. After it will be revealed the usefulness of the FFT to convert time domain measurements to frequency domain signals for detailed analysis. Initial measurements have been conducted and similarities between pressure and acceleration measurements will be investigated in the frequency

range of interest (< 6 kHz). Lastly, preliminary flow visualization results are available to connect flow regimes to noise measurements.

ACKNOWLEDGMENTS

I wish to thank my advisor, Professor Stefan Elbel for his guidance and care during my years at the University of Illinois at Urbana Champaign. His advices and suggestions were essential to the development of this work.

I had the pleasure to cooperate with a wonderful team at the ACRC and with many professors, who were always able to give me the teachings and suggestions.

Amongst all my friends, I would like to specifically thank Yishu for helping me edit the thesis, and Paolo for his much-needed encouragement during these two years.

Finally, I would like to express my deepest gratitude and reverence for my family back in Italy. Without the unconditional and continuous love and support of my father Mauro and mother Luisa, this thesis could never have been completed. My girlfriend Alessandra, was always there during these two exciting, and sometimes hard, years. Her love and understanding were the only things driving me forward at the loneliest of times.

TABLE OF CONTENTS

1. INTRODUCTION

The topic of the following thesis will be the description of my experience as Research Assistant at the Air Condition & Refrigeration Center (1206 W Green St – 61801 IL, US) from here on reported with his acronym of ACRC.

In this introduction the objectives of the research, the objectives of the thesis, and the starting point, also known as state of the art, will be discussed.

1.1 OBJECTIVES OF THE RESEARCH

A cultural context where acoustic technologies are rapidly growing, and the demand for comfort is increasing exponentially, were all catalysts for the birth of this research project. In particular, noise is one of the major concerns that many companies want to come against, especially in the HVAC & Refrigeration sector. Indeed, noise coming from air conditioning systems can be perceived as very annoying. Thus, one of the main reasons this research is conducted is to increase consumer comfort in all applications where air conditioning noise and in particular noise coming from the expansion device, is perceived as very disturbing. While another motivation to conduct research in this area is to improve the understanding of flow induced noise.

The targets for this research are to:

- Identify new ways of reducing noise emitted from expansion devices in HVAC & Refrigeration systems;
- Relate flow regime to flow induced noise using visualization;

- See if all refrigerants behave in the same way;
- Correlate flow pressure fluctuations and structure accelerations with noise measured by microphones;
- Understand how to measure flow induced noise and the relative measurement techniques.

For the purposes of this thesis, this last target will be the main focus.

1.2 OBJECTIVES OF THE THESIS

This thesis will not talk about all the above-mentioned research targets, but rather, it will focus on measurement techniques, analysis methods, and some general theory about expansion devices and noise.

The reason behind this choice is not only practical, but also in case future scientists who would like to continue this research, they can use this thesis as a stepping stone to learn how to create this system and have the basis to delve deeper into the problem at hand.

1.3 BACKGROUND (or STATE OF THE ART)

Work in this field has been previously conducted by multiple scientists, here some of the most relevant works for this research will be reported.

- Going back to 1999 Singh et al. [8] conducted a research related to flow noise in the expansion device:

- Measurements and predictions of flow noise propagated to the environment from expansion devices (using R134a) were developed.
- Their measurements are based on the Two-Microphone Technique shown below (fig 1.1).

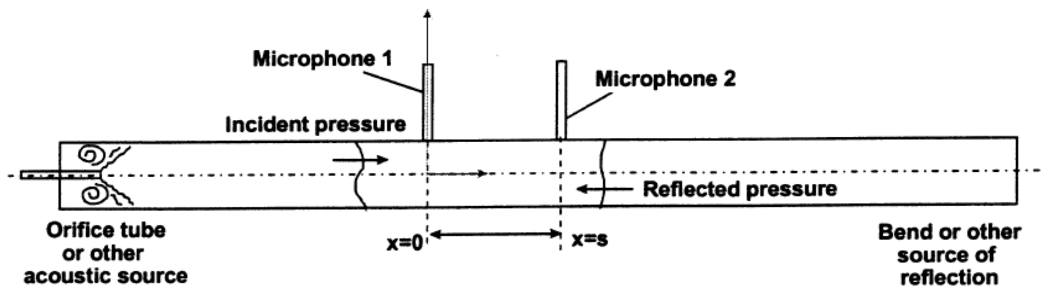

Figure 1.1: *Two Microphones Technique [8].*

- Hirakuni *et al*., 2004, [12] studied noise reduction with porous metal for refrigerant two-phase.
- The results show that porous metal upstream of the orifice improves homogenous mixing vapor and liquid downstream
- It reduces pressure fluctuations of two-phase flow; therefore, noise reduction is achieved.

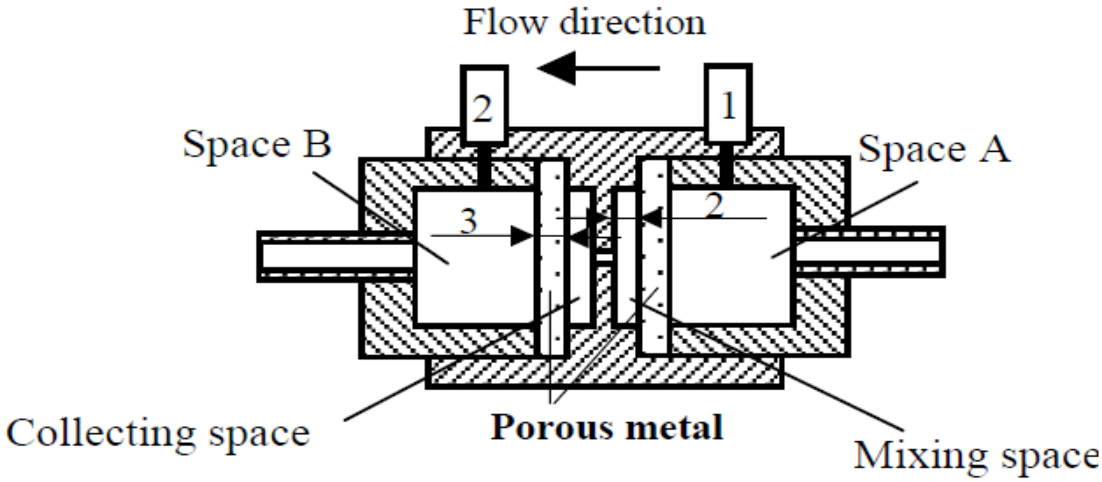

Figure 1.2*: Setup Hirakuni et al. research [12]*

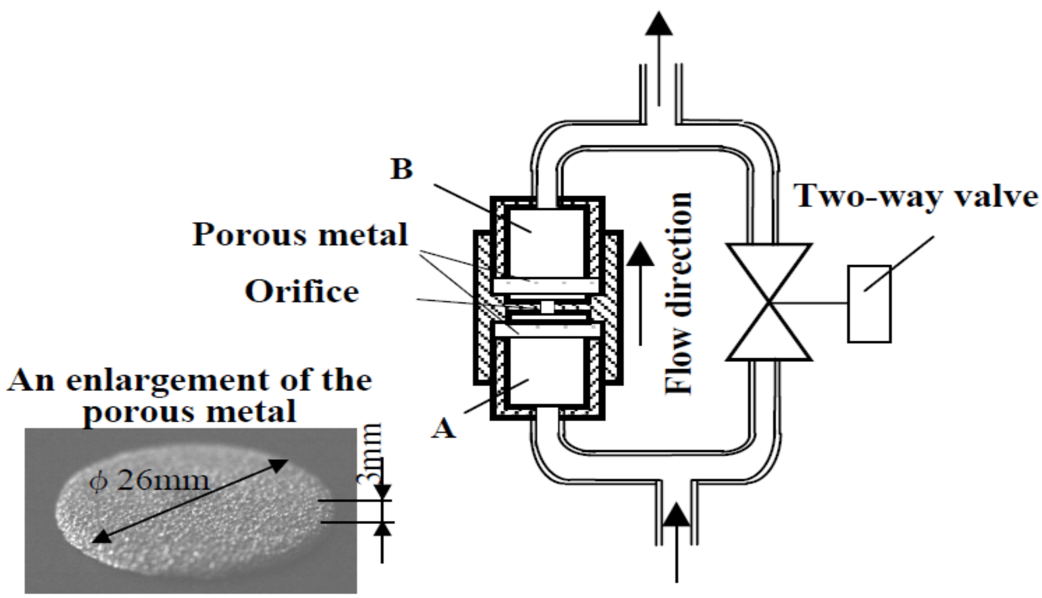

Figure 1.3: *Porous Metal [12].*

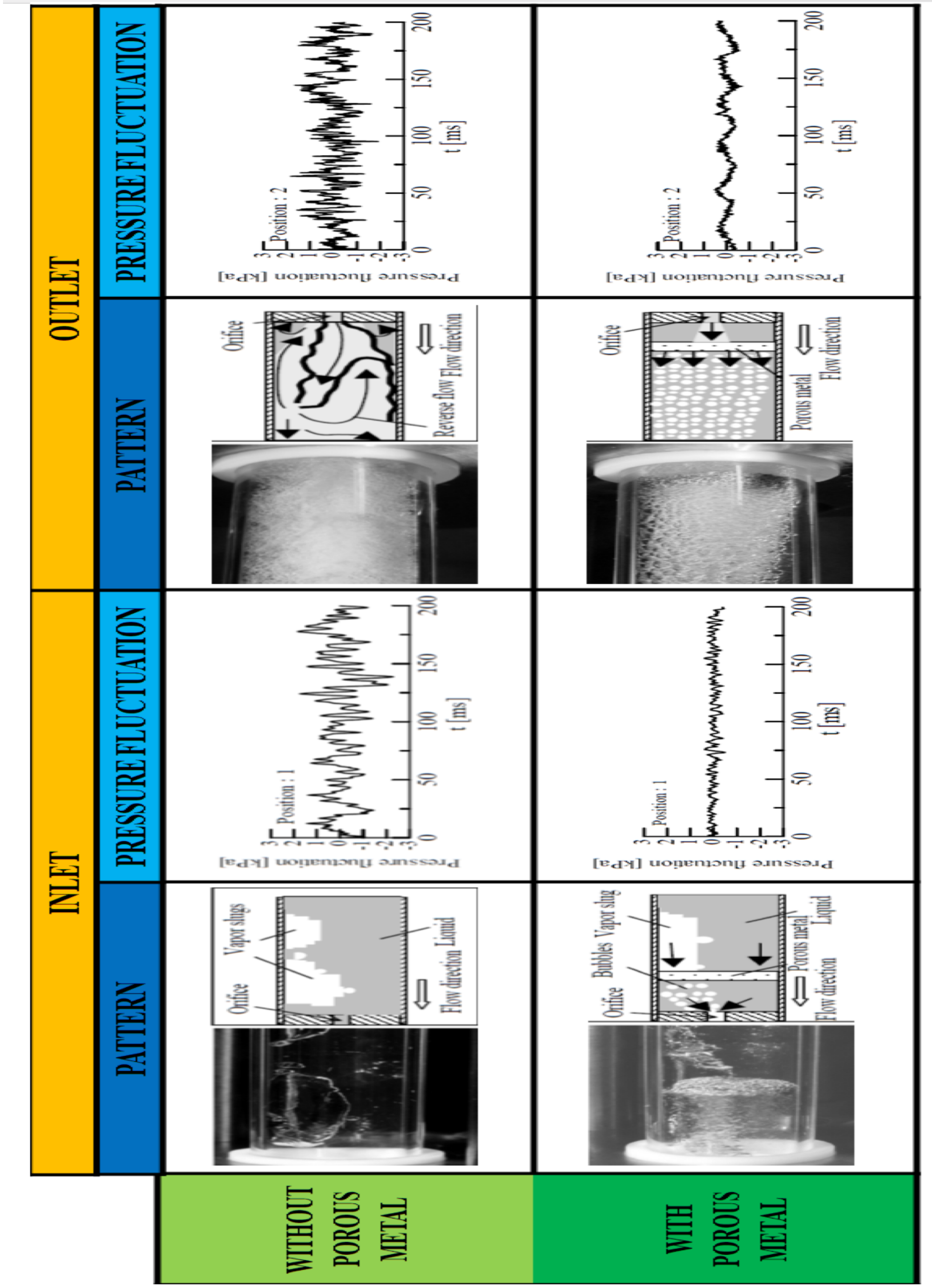

__Table 1.1__: Experimental Conditions Hirakuni et al. research [12].

- Tannert T. *et al.,* 2016, [15] found a coincidence in flow pattern fluctuations and changing noise levels. Reason: improperly designed cap tubes cause mass flow rate fluctuations leading to noise generation (not enough subcooling at cap tube inlet).
- Liu Z. *et al*., 2015, [11] proposed a solution to gurgling noise, in particular, they removed sharp 90-degree bends to reduce the intermittency of flow entering TXV.

2. EXPANSION DEVICES OVERVIEW

2.1 INTRODUCTION

The expansion device is a device able to generate a rapid and significant pressure drop – with an increase in volume - in the working fluid of a system.

The most common applications of the expansion device can be found in the HVAC and refrigeration systems, where it has three main functions:

- A thermodynamic function to expand the liquid refrigerant from condenser pressure to evaporator pressure;
- A control function to meter the volume of refrigerant entering the evaporator, thus respecting its cooling capacity;
- A capacity to ensure that superheated refrigerant is exiting the evaporator.

Even though the degree of decrease in temperature depends largely on the type of device chosen, a common feature for every expansion device is the use of a small diameter tube or orifice section to restrict the working fluid's flow and create frictional losses. There also exist more complex expansion valves that can use a variable flow area in addition to small diameter flow passages. To help increase the pressure drop and regulate the refrigerant flow, many stratagems are used for example, the fluid usually ends up in larger diameter tube, which creates exit flow losses.

Currently, even though there are many different types of expansion devices, in Air-Conditioning Systems only two main categories of throttling devices are used, the Thermostatic Expansion Valve and the Fixed Orifice Valves. In order to determine which type of valve is the most appropriate, the pressure drop across the evaporator is used in order to make a conclusion.

The pressure drop across the valve may cause some of the refrigerant to flash, resulting in a two-phase low temperature mixture after the valve. Many times this vaporization may be assumed to not occur until after the fluid has passed through the valve, as the liquid inside is momentarily in a metastable condition, and liquid flow equations apply [1]. This occurs particularly for valves where the pressure drop occurs quickly, across a short flow distance; the expansion is approximately isenthalpic and adiabatic (as it will be explained in the next paragraph).

2.2 PROPERTIES

The properties of expansion devices are thermic, fluid and mechanical. There are several common assumptions that are made regarding the energy analysis of throttling devices:

- No significant heat transfer due to the small dimension of these pieces of equipment:

$$\dot{Q} \cong 0.$$

- Expansion devices are properly shaped and no work can be transferred in or out:

$$\dot{W} = 0.$$

- As a fluid passes through a throttling device, it does not experience a significant change in its velocity, which makes the change in kinetic energy insignificant:

$$\Delta \mathrm{Ke} \cong 0.$$

- The change in potential energy is usually very small $\Delta \mathrm{Pe} \cong 0$.

  So, the energy balance for throttling devices is:

$$\mathrm{h}_2 \cong \mathrm{h}_1$$

When this assumption is made, we call it throttling process. It is a process in which there is no change in enthalpy from state one (before the expansion device) to state two (after the expansion device) and that is adiabatic (an almost realistic assumption).

This theory can be confirmed by the observation that usually $\mathrm{vel}_1 < \mathrm{vel}_2$ (velocities), while $\mathrm{P}_1 > \mathrm{P}_2$ (pressures). Keeping in mind that $\mathrm{h} = \mathrm{u} + \mathrm{Pv}$ (v, specific volume), if pressure decreases then the specific volume must increase to maintain enthalpy constant; however, because mass flow is constant, the change in specific volume is observed as an increase in gas velocity, and this is verified by the above specified observations.

2.3 CHARACTERISTICS OF MAIN EXPANSION DEVICES

Following is a brief description of the technical characteristics of the main categories of expansion devices used in refrigeration systems. The focus will be on manufacturing and technical characteristics, and the working principle of the expansion devices. Subsequently the uses/applications and the noise impact of the said devices will be mentioned. In the coming chapters we will delve deeper into noise.

2.3.1 CAPILLARY TUBE

a) Manufacturing and technical characteristics:

One of the most common expansion devices used in refrigeration and air conditioning systems is the capillary tube.

Made of copper it has a very simple and inexpensive design characterized by a very small internal diameter (about 0.5 – 2.3 mm) and a length varying from 1 to 6 meters, which create a pressure drop in the working fluid. Due to the important length of the tube, it is coiled to several times to occupy less space.

One disadvantage of this device is that it allows optimal performance for only one set of operating conditions. Indeed, since the orifice tube is made of a simple copper tube, it is not possible to adjust it to change load conditions – this generates inefficiencies not present in a system with variable refrigerant flow.

It is also Important to notice that due to the small cross-section area of these kind of tubes, they are susceptible to clogging caused by foreign matter, so more attention must be paid while charging the system with the working fluid.

b) Working principle:

Once the refrigerant reaches the capillary tube, the pressure drop happens mostly because of the small opening on the tube rather than the orifice.

The decrease in pressure of the working fluid depends on the diameter and length of the capillary tube. The longer the tube and the smaller the diameter the bigger the pressure drop and decrease in temperature of the refrigerant will be.

The capillary tube design allows pressures to equalize during the off cycle, due to this reason when the compressor restarts there will be a small load on it (thus the motor driving the compressor can be one of low starting torque [1]). Also, due to this reason attention must be paid to not over-charge the refrigeration system with the refrigerant.

c) Noise impact:

In terms of noise generation these devices are not ideal. This is because the frictional pressure drop that refrigerant vapor experiences during flow through capillary tube expansion devices leads to reduced vapor densities and corresponding flow velocity increases. Chocked refrigerant vapor flow and shocks may also occur.

Since vapor flow noise generation is strongly related to flow velocity, this last one is not desired to be high at the exit of the expansion device [15].

d) Uses and applications:

As stated before one common application of the capillary tube is in HVAC and refrigeration systems, wherever there is no need in changing the operation conditions; particularly, it is used in domestic refrigerators, deep freezers, water cooler and air conditioners.

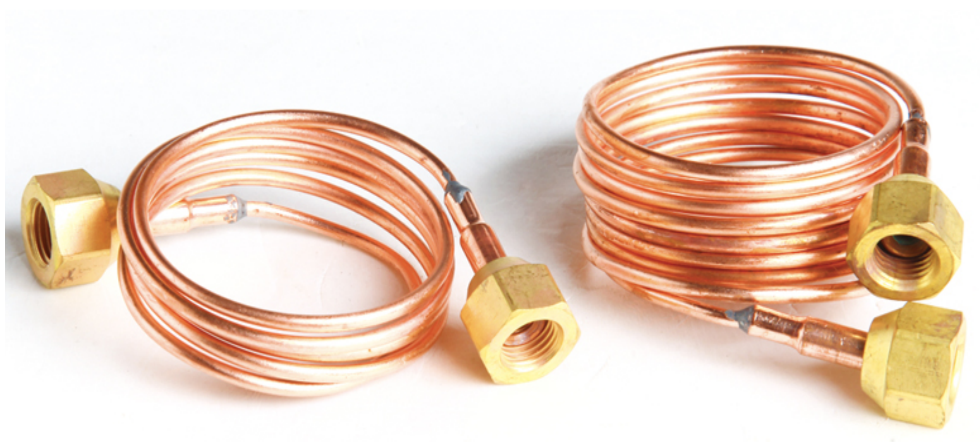

Figure 2.1: *Capillary tube [2].*

2.3.2 ORIFICE PLATE (OR RESTRICTION PLATE)

a) Manufacturing and technical characteristics:

The restriction plate is a thin plate or a series of thin plate sections with small holes for the fluid to flow through. There are three main types of restriction orifice plates:

1. Single stage restriction orifice (fig 2.2);
2. Single stage multi-hole restriction orifice (fig 2.3);
3. Multi-stage restriction orifice plate assembly (fig 2.4).

They are usually manufactured in stainless steel, Monel, Hastelloy B or C, Titanium, or Teflon.

Orifice plates can be used in series to get a stepwise pressure drop without requiring a long expansion flow distance or a lot of space for the device itself.

Some disadvantages include the inadequacy to respond to changing load conditions and the possibility to be clogged by debris due to the small diameter of the device. This is due to the fact that there is no pressure drop coming from frictional loss, and so the orifice needs to be smaller compared to other expansion devices which incur frictional losses.

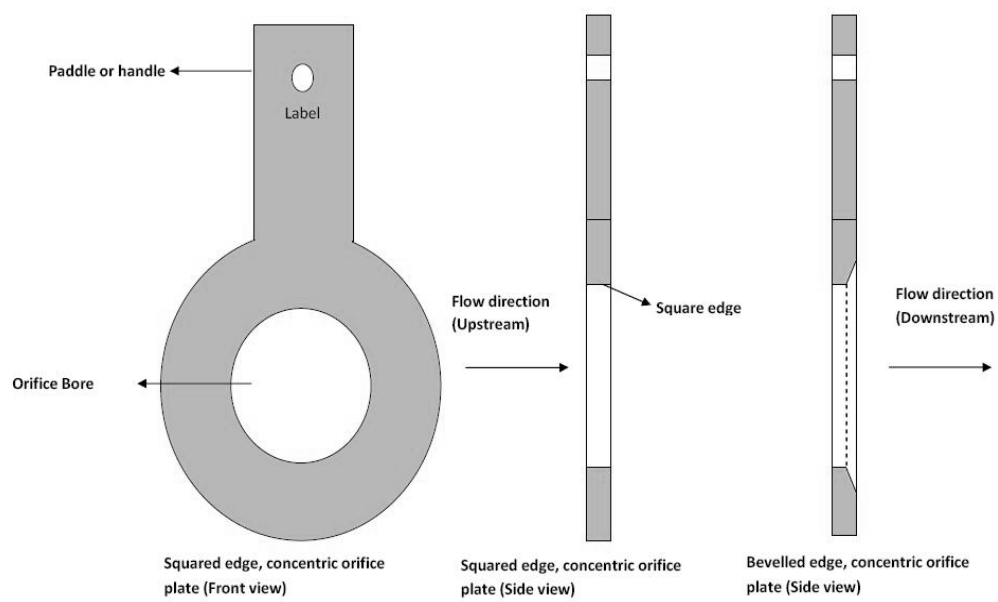

Figure 2.2*: Single Stage Restriction Orifice.*

Figure 2.3*: Single Stage Multi-Hole Restriction Orifice.*

Figure 2.4*: Multi-Stage Restriction Orifice Plate Assembly.*

b) <u>Working principle:</u>

This device generates a pressure drop suddenly contracting and expanding the fluid, respectively, at the inlet and outlet of the plate. The magnitude of pressure drop strongly depends on the refrigerant used, the system configuration (i.e. outer tube diameter), and the orifice diameter.

c) <u>Noise impact:</u>

The noise coming from cavitation in the flows can often generate high levels of noise. Not to mention the strong vibrations of the system in which they are installed. It can be a serious issue for industrial applications.

d) Uses and applications:

It is worth to mention that one of the orifice plate common usage includes measuring flow rates, in particular for single-phase fluids.

Another application, more in line with the focus of this thesis, is found as expansion device in air conditioning and refrigeration systems, where it is also called restriction plate.

The advantage of the capillary tube is that it is a cheap device, due to the relative simplicity of design and realization.

2.3.3 ORIFICE TUBE

a) Manufacturing and technical characteristics:

Similar to the previously analyzed expansion devices, the orifice tube has a simple design, indeed it also characterized by the fact of not having any moving parts.

From figure 2.5 it is possible to appreciate the various components of this broadly used expansion device:

- The Orifice, the core of this expansion device through which the refrigerant flows, is made of brass. To better understand the working principle, note that the tube is much smaller than a capillary tube, but still longer than the orifice plate.

- The O-Ring (made in rubber) is used to seal the high side from the low side.
- The Inlet Screen is used to protect the orifice tube from getting clogged with debris present in the system.
- The Diffuser screen, used to reduce noise.

Finally, as for the other devices till here described, the orifice tube is not adjustable; so, it is usually used when the refrigeration needs are steady.

Figure 2.5: *Typical Orifice Tube from GM (General Motors).*

b) <u>Working principle:</u>

Regarding the working principle, the refrigerant flows through a tube (the orifice) with a small flow diameter, which due to the above-mentioned length consents to little frictional loss.

c) <u>Noise impact:</u>

Under normal working conditions, the noise generated by the orifice tube is partially mitigated by the diffuser screen; however, when debris clogs the hole, the noise generated from this expansion device can be very strong. Therefore, for long term usage without maintenance, the orifice tube is not the optimal solution in terms of noise reduction.

d) <u>Uses and applications:</u>

The orifice tube was originally introduced on GM applications in the 1977 vehicle model year [3]. Nowadays it is very popular and it is used by several manufacturers, especially in the automotive industry. The biggest disadvantage of the orifice tube is that it tends to get plugged up by the debris that may travel in the refrigeration system; once the orifice is clogged, the passage of oil -mixed to refrigerant- throughout the system is restricted and can lead to compressor failure.

2.3.4 VARIABLE FLOW EXPANSION DEVICES

Up until this point only fixed area restrictors have been treated, now a different kind of expansion device will be approached. Specifically, variable flow expansion valves. These are more complex expansion devices that utilize diverse metering systems to govern flow through an orifice section. Valves can be controlled using mechanical or electrical methods, based on the desired outcome.

The most relevant benefit given by expansion valves is the ability to respond to changing load conditions in order to meet performance goals or achieve higher system efficiencies.

Among the most used expansion valves we can distinguish three main types:

- Constant Pressure (or Automatic) Expansion Valves;
- Float Expansion Valve;
- Thermostatic Expansion Valve.

Next is a description of these types of variable expansion devices.

2.3.4.1 AUTOMATIC EXPANSION VALVE

a) Manufacturing and technical characteristics:

Looking at fig 2.6 it is possible to appreciate the manufacturing details of the automatic expansion valve.

Externally there is a metallic body with two openings through which the refrigerant flows from the condenser to the evaporator, while internally all the mechanisms are delineated.

Starting from the upper part, it is possible to see the adjustable screw that controls the pressure of the spring and so the metallic diaphragm. This last piece of equipment is connected to the orifice –composed by the seat and the needle- through a push rod.

b) <u>Working principle:</u>

The main principle of the constant pressure valve, is to automatically maintain a constant pressure inside the evaporator regardless of the other variables of the system and of the fluid load inside the evaporator.

When the diaphragm moving down, the needle also moves down away from the seat, leading to the valve opening. This whole system is activated by the adjustable spring, which together with the atmospheric pressure puts the diaphragm under compression. On the other side, below the diaphragm, there is the working fluid at the same pressure of the evaporator. That itself puts the diaphragm in compression, but this time moving it upward.

As a result, if the pressure at the evaporator increases to a higher level than the pressure coming from the spring (previously regulated through the adjustable screw), the needle will move up in the seat direction, closing the valve.

Under normal operating conditions of the system, there will be a balance between the two pressure, allowing for an opening of the valve, so that there will always be refrigerant flowing through it.

As briefly mentioned above, it is possible to regulate the evaporator pressure by simply regulating the adjustable screw and so the spring.

From a safety point of view, it is worth noting that this device is particularly suited for all the applications where pressure of the system is a sensible factor. Furthermore, when the system is switched off there will be a certain amount of refrigerant trapped in the evaporator, which due to its pressure, will act on the diaphragm and consequently closing the valve. Eliminating any risk of flow inversion. This action will stop as soon as the plant will restart.

c) Noise impact:

Since noise is directly proportional to pressure, it is intuitive to expect a lower steady state noise for automatic valves compared to other variable flow expansion devices. However, with this device noise can be created indirectly. Indeed, it is well known from previous researches, that due to the impossibility to regulate the load on the evaporator, a higher noise can be perceived from this last piece of the plant.

d) Uses and applications:

The main uses of the pressure-controlled valve are to maintain the evaporator temperature and pressure at a specific point, to control humidity, and prevent freezing in water coolers.

As mention before, it is also important from a safety point of view of the plant, to protect the compressor from overload due to the high suction pressure.

Finally, this valve is particularly suitable whenever the work is done with small systems, because in these cases a critical charge of the refrigerant is feasible to prevent the fluid from flooding out of the evaporator, under extreme pressure conditions.

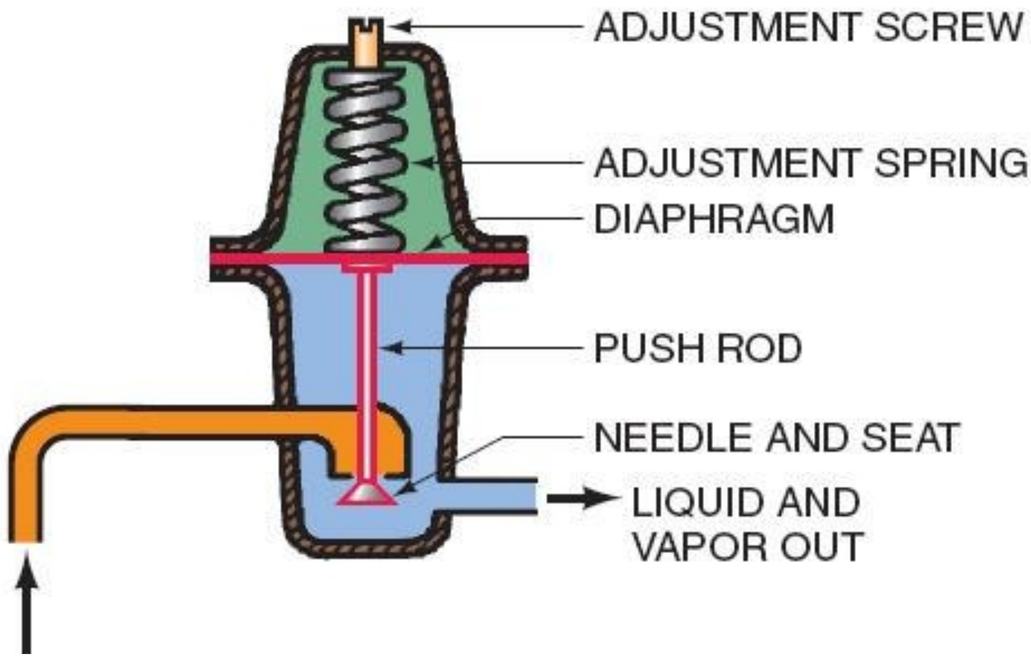

Figure 2.6: *Automatic Expansion Valve [18].*

2.3.4.2 FLOAT EXPANSION VALVE

There are two main types of float expansion devices that will be described here, low and high pressure (side) float valves.

a) Manufacturing and technical characteristics:

The float expansion valve helps maintain liquid refrigerant in the evaporator at a determined level, irrespective of the pressure and temperature in it.

The distinguishing technical characteristic between these two types of valve, is that low pressure float valves are placed in the evaporator (from here the name, since it is the low-pressure side). While, the high side valve, as the name suggest, is placed between the condenser and the evaporator, where the pressure is higher.

These valves (as shown in fig 2.7 and 2.8) are composed by the float ball, the float arm, the valve, and the seat of the valve, these last two forming the orifice opening. The needle valve is connected to the float ball by means of the float arm, which commands the opening and closure of the orifice.

b) Working principle:

The working principle of low side valves is like the ones used for water tanks, where the valve opens when the level of refrigerant is too low in the evaporator. While, for high side valves, the float valve is actuated whenever the refrigerant increases in the chamber.

c) Noise impact:

Due to extensive use in big refrigeration plants, specifically at the fluctuating float and the continuous change of pressure at the inlet and outlet, the float expansion valve is expected to be very noisy. However, this noise is usually not perceived as a big issue since, as stated above, it is of large use in industrial applications, where the majority of noise is usually coming from the compressor and other big machines.

d) Uses and applications:

This type of expansion valve is often used in large industrial refrigeration plants as a cooling system for the entire process.

Specifically, the low side float valve is usually used in flooded type refrigeration plants, where it is required to lower the temperature of a large amount of substances. While the high side expansion valve is mostly used for centrifugal chillers.

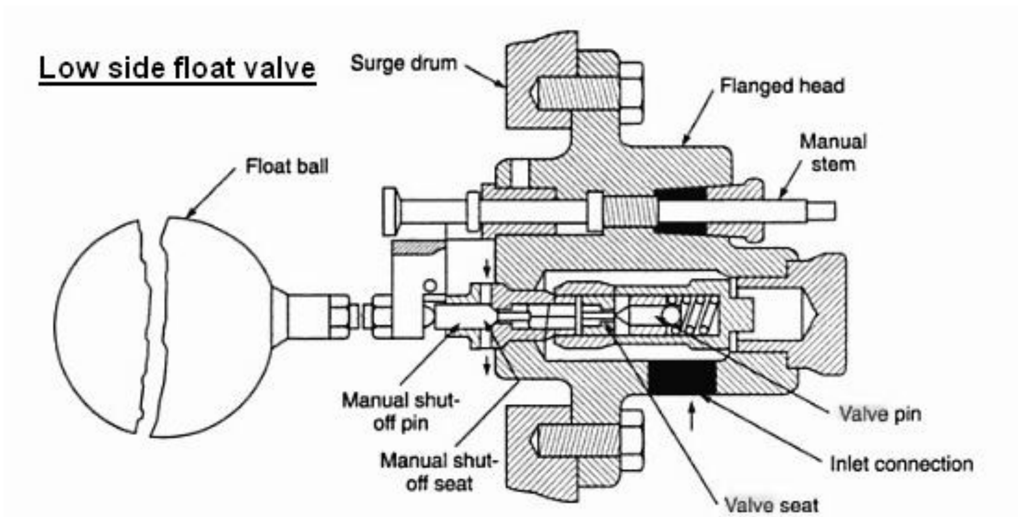

Figure 2.7: *Low Side Float Valve* [4].

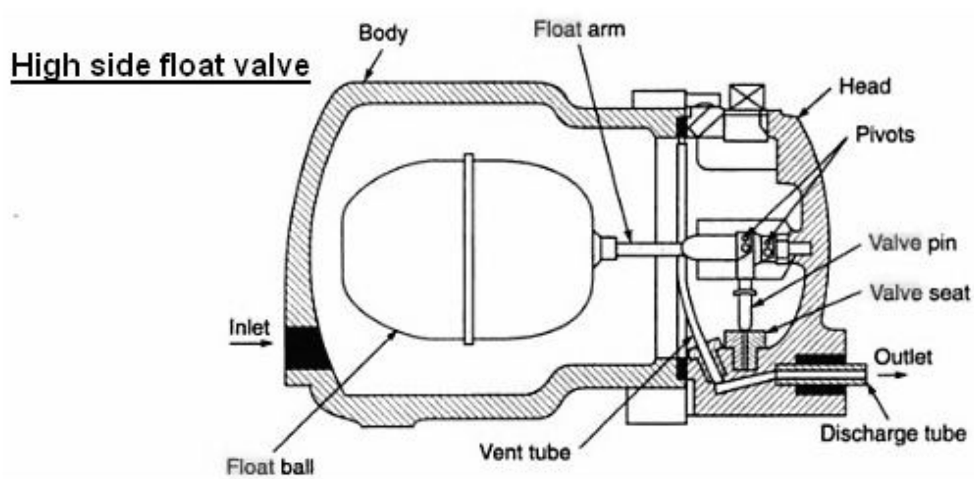

Figure 2.8: *High Side Float Valve* [4].

2.3.4.3 THERMOSTATIC EXPANSION VALVE

a) Manufacturing and technical characteristics:

Thermostatic expansion valves, also called TXV, are usually electro-mechanically controlled devices.

TXVs are composed by three main parts (as shown in fig. 2.9), the capillary tube and bulb, the power head, and the body.

b) Working principle [17]:

TXV maintains a certain magnitude of superheat of the suction gas leaving the evaporator. They can be controlled electromechanically or by a diaphragm integral to the valve. Valve position, and hence flow, is regulated in proportion to the rate of evaporation in the evaporator. This prevents liquid refrigerant droplets to enter the compressor and it also maximize the effect of the refrigerant.

c) Noise impact:

On average, it is probably the less noisy expansion valve. However, specific data for this work are not available.

d) Uses and applications:

Thermostatic expansion valves are by far the most popular type of expansion device used in modern commercial refrigeration systems.

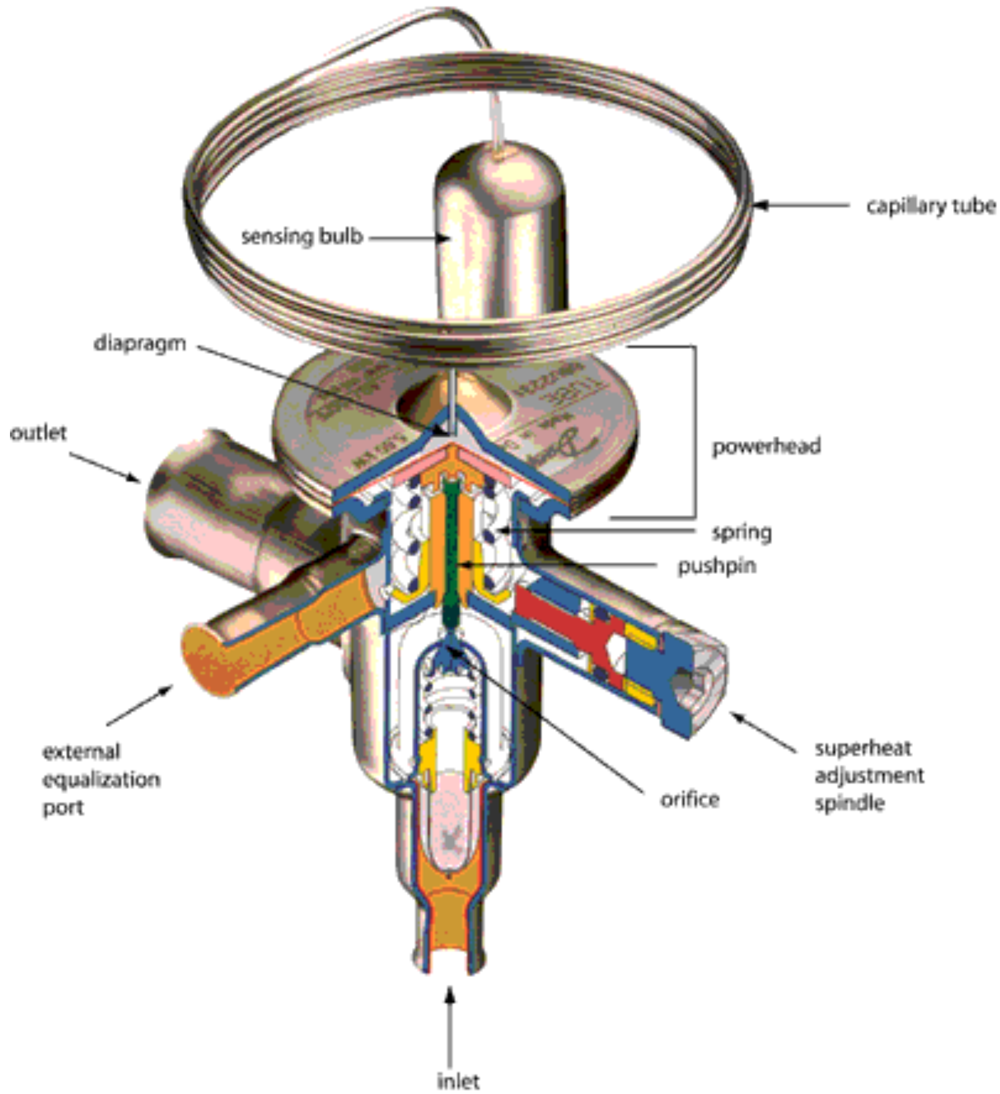

Figure 2.9 : *Thermostatic Expansion Valve* [5].

3. NOISE THEORY AND QUANTIFICATION METHODS

3.1 NOISE THEORY

Following is a brief introduction to the concept of noise, with an introduction to the specific problems in HVAC & refrigeration systems.

3.1.1 INTRODUCTION

In HVAC & refrigeration systems noise is considered one of the biggest issues, not to mention the excessive vibrations often associated with it which can also cause structural problems. In particular, for all those applications where the noise originating from other sources has been reduced or eliminated, it becomes an even more important topic. For example, thinking about the automotive sector, immediately electrically driven vehicles come to mind, where the noise has been reduced to an extent that makes the air conditioning disturbance even bigger. In this case, even the most moderate sources of noise can be perceived as very disturbing to the user. Another example can be found in residential air conditioning systems. Every unit pipe condenses refrigerant to each room. After that the refrigerant is expanded and passed through each room's evaporator. Residents complain because they hear an annoying noise caused by the expansion device and the other components.

3.1.2 THE CONCEPT OF NOISE AND SOUND

Noise is unwanted sound judged to be unpleasant, loud or disruptive to hearing. Sound is nothing else than waves, generally it consists of a source, a path, and a receiver. As above stated, noise can be perceived differently based on different receivers, hence, a strong emphasis is given to noise characteristics such as loudness, sharpness, tonality, and modulation. From a physics standpoint, noise is indistinguishable from sound, as both are vibrations working through a medium such as air or water. The difference between the two arises when the brain receives and perceives a sound [6 and 7]. Thus it is clear that noise is a simple, although complex, concept. However, for this research, noise is considered as the sound that a human can hear. From now on the word sound will be used in place of noise as synonym.

3.1.3 CHARACTERISTICS & MAIN PROPERTIES

The purposes of this work is to identify a reference regarding the level at which sound would not be perceived as disturbing. Looking at the classical theory of noise, it is found that the Absolute Threshold of Human Hearing (ATH) is of 20 μPa (0 dB) at 1atm, 25°C and 1 kHz (figure 3.1). At this level, the average human cannot perceive any noise.

It is interesting to notice how the ATH is given as a combination of sound, pressure, and frequency. This is due to the fact that human hearing perceives loudness as a combination of these last two.

Indeed, based on norm ISO 226:2003 (fig 3.1), equal loudness (measured in phon) can occur for different combinations of frequency and sound pressure.

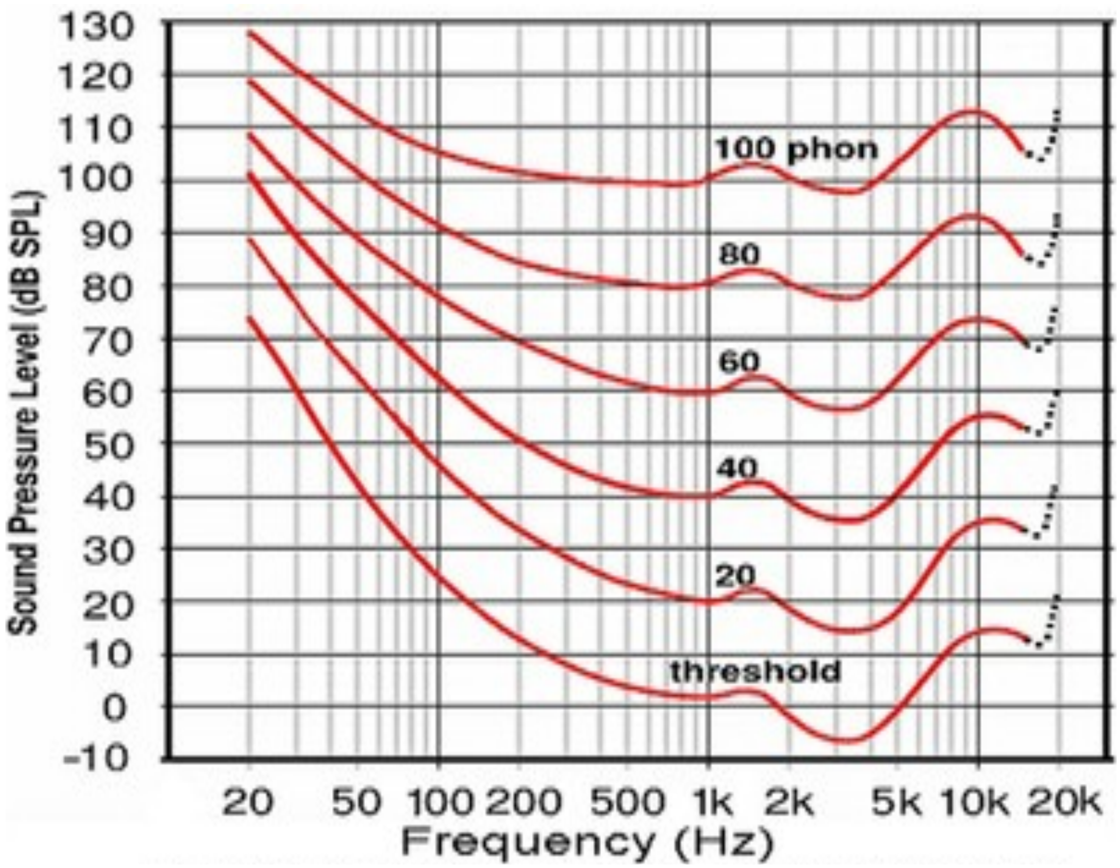

Figure 3.1: *Equal Loudness Contours from ISO 226:2003.*

The other properties of sound apart from frequency (inverse of the wavelength) and sound pressure level (SPL), are the amplitude of the sound wave, speed of sound, and direction.

Sound pressure level [dB] is defined as:

$$SPL = 10\log_{10}\frac{p^2}{p_{ref}^2} = 20\log_{10}\frac{p}{p_{ref}}$$

Where *p* is the rms (root mean square) of sound pressure, and p_{ref} is the sound pressure reference. Commonly used references for air and water are $20\ \mu Pa$ and $1\ \mu Pa$ respectively.

Regarding frequencies and wavelength, people can only hear from 20 Hz up to 20 kHz, which can be translated as 17 m to 17 mm wavelengths in the air at standard temperature and pressure.

The sound's wave speed is defined as:

$$v = \lambda f$$

where λ is the wavelength and *f* is the frequency.

Finally, it is worth describing sound speed which varies depending on the path it passes through and temperature. For example, speed of sound through the air at 0 °C is 331 m/s, while at 20 °C it is of 343.8 m/s. Sound speed through the air varies linearly with temperature following the equation:

$$a(T) = 331.45 + 0.62 * T$$

3.1.4 NOISE SOURCES

Noise can come from different sources, two of them are mechanically driven components and the refrigerant flow. Some of the elements that can contribute to the first kind of noise are compressors, pumps and, air moving equipment. However, for this thesis we will focus mostly on the noise caused by the refrigerant flow which specifically comes from the expansion device.

As seen in the previous paragraphs, vibrations or flow fluctuations become noise once they reach a fluid path, a medium, through which they can propagate; usually the air. Figure 3.2 shows this concept.

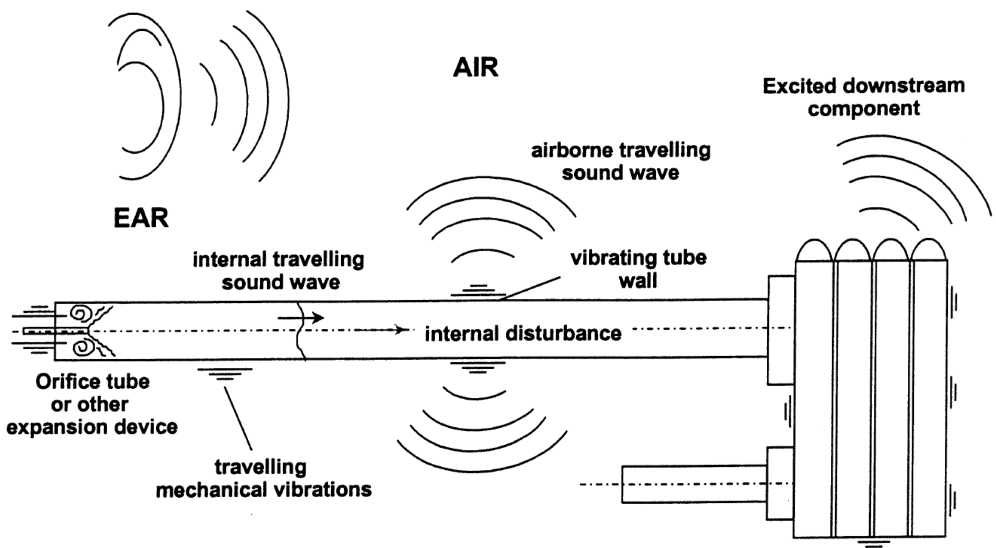

***Figure 3.2**: Expansion Valve Noise Generation and Propagation* [8].

In the above figure, it is possible to note the airborne sound wave which results from structure-borne noise.

Structure-borne noise is the result of flow induced and mechanically excited noises. The first one is generated by local pressure fluctuation in the refrigerant flow, while the second is caused by the vibrations (continuous or periodical) of the structure's mechanical components.

The following paragraph will explain the fundamentals of flow induced noise in deeper details.

3.1.5 SOUND WAVES IN TUBES

Based on the definition of Webster's dictionary [9], a wave is 'a disturbance or variation that transfers energy progressively from point to point in a medium and that may take the form of an elastic deformation or of a variation of pressure, electric or magnetic intensity, electric potential, or temperature'.

This definition applies also to sound that can be defined as pressure waves that move through an elastic medium (air or fluid) at a specific speed. Sound waves are created by pressure fluctuations that can be caused by both vibrating sources or fluid flow.

It is important to understand that the medium through which the wave travels may experience some local oscillations as the wave passes through, but the particles in the medium do not travel with the wave. The disturbance may take any of several shapes, from a finite width pulse to an infinitely long sine wave [10].

Following is an introduction to plane waves that will be useful to better understand paragraph 3.1.5.2.

3.1.5.1 PLANE WAVES [8]

As above stated sound is a function of frequency, what is new however, is that in most cases sound can propagate from a single source at different frequencies.

In a duct, the waves at lower frequency propagate as plane waves. The wave fronts are surfaces perpendicular to the direction of propagation with uniform sound pressure. Figure 3.3 shows the particle velocity of a plane wave.

Now that the plane wave has been defined, it is important to note that every mode of a wave has a frequency over which the wave can propagate in different shapes. This specific frequency is called first cutoff frequency. For a duct of circular cross-section, the first cut-off frequency can be estimated by the following equation. Where d_{in}[m] is the tube's inner diameter and *c* is the speed of sound in the working fluid [m/s].

$$f_{cutoff} = 0.586 \frac{c}{d_{in}}$$

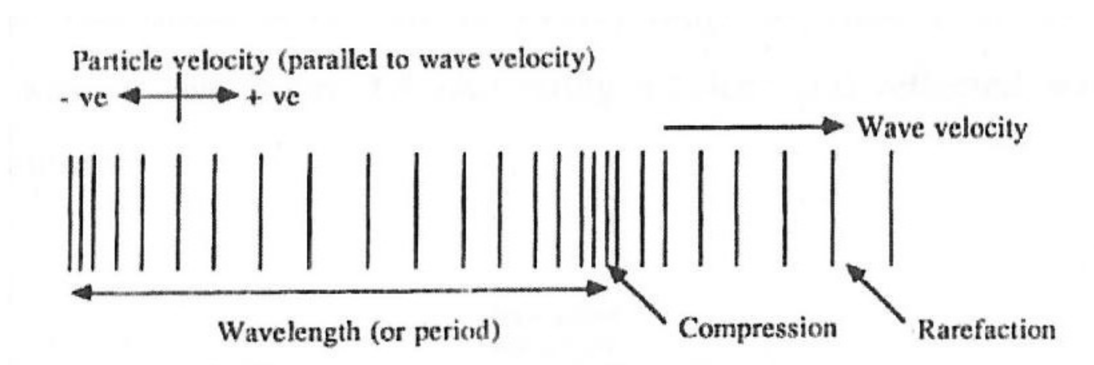

***Figure 3.3**: Compression and Rarefaction of a Sound Plane Wave* [16].

If due to any obstruction the plane wave is disturbed, it will decay and return to a planar shape in 2-3 duct diameters.

Finally, it is worth underlining that, due to frictional, turbulence, and heat transfer losses, as the wave travels though the duct the sound pressure will slowly decrease.

3.1.5.2 REFLECTION AND STANDING WAVES [8]

Reflection of sound waves is a common phenomenon to happen in tubes. Reflection of sound waves can be particularly common whenever a change in an area or a medium happens. For example, it can happen at the exit of the flow pipe or due to a disturbance in the downstream flow path.

For noise measurement, this phenomenon must be carefully considered. Indeed, once the reflected waves interact with the incident waves, the noise measurement will be compromised.

Following is an illustration (figure 3.4) of how incident and reflected waves interact with each other in a tube.

A particular case is when the incident and reflected waves are plane waves. For this specific case it is possible to use the following equation to describe mathematically the standing wave pattern in the flowing fluid:

$$P(f,x) = P_i(f)e^{-j\frac{2\pi f x}{c(1+M)}} + P_r(f)e^{j\frac{2\pi f x}{c(1-M)}}$$

Where *P(f,x)* is the Fourier transform of the measured acoustic pressure [Pa] at location *x* [m], $P_i(f)$ is the Fourier transform of the incident acoustic pressure at *x*, $P_r(f)$ is the Fourier transform of the reflected acoustic pressure at *x*, *f* [Hz] is the frequency, *c* [m/s] is the speed of sound, and M is the Mach number of the flow.

Due to the dependence of the acoustic signal on this phenomenon, it is important to understand that for a given incident acoustic signal, a different set-up will give in general different measurements of sound pressure. Hence, different from the ones obtained in this work; which is why it is vital to reduce as much as possible the presence of reflected waves, or at least being able to account for them in the acoustic measurements.

This will be clearer in the next chapter about the experimental facility.

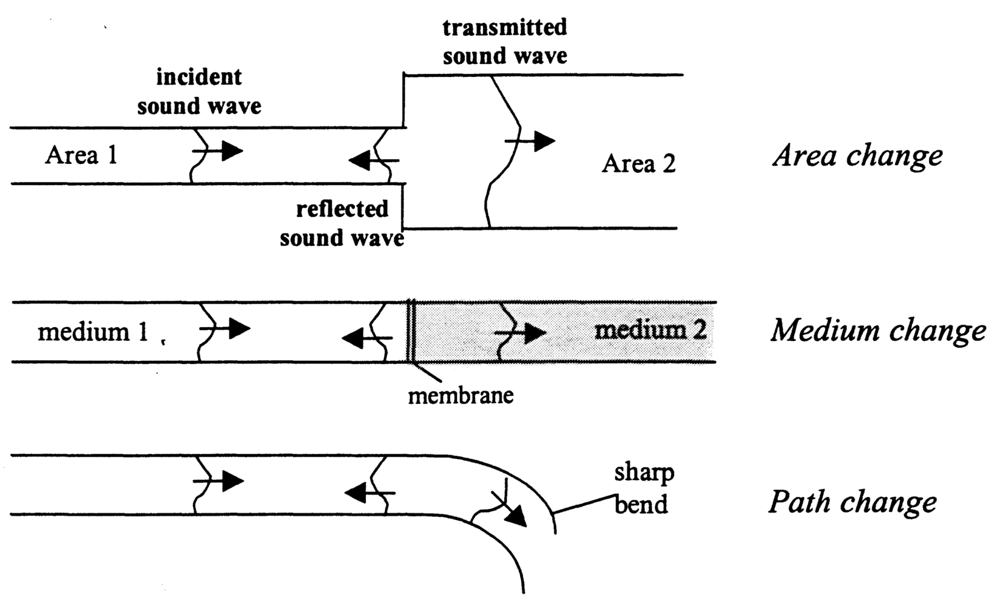

Figure 3. 4: *Possible Sources of Reflection of Sound Waves in Duct Flow [8].*

3.1.6 NOISE TYPES & PATTERNS

Among many classifications, noise can be also categorized in two different ways: types of flow-induced noise, and patterns. Regarding pipe-flow, the first category can be classified as popping, gurgling and hissing. While concerning patterns, the two main types are intermittent and continuous noise. For example, popping and gurgling are intermittent noise, while hissing is continuous.

It is also curious to know that based on some recent studies (e.g. Hirakuni et al., 2014, Chung et al., 2012) often noise is perceived as more disturbing when intermittent as opposed to continuous.

Finally, analyzing the three main types it is possible to see that:

- Hissing: is often steady noise typical for expansion valves, which is caused by high velocities during the expansion of the refrigerant from high to low pressure.
- Gurgling: is a transient noise either in the expansion valve or in adjacent piping. It usually happens for short periods of time when the system is running. It also happens when the system is turned on or off. This is due to vapor bubbles travelling through liquid.
- Popping: usually happens in the expansion valve due to collapsing vapor bubbles.

3.2 QUANTIFICATION METHODS

In order to quantify flow induced noise it is possible to use several different ways. There are both direct and indirect methods.

Because the noise is generated in the expansion valve by the working fluid (i.e. the refrigerant), a direct method for measurement, is one that includes measuring the pressure fluctuations directly into the refrigerant. While an indirect method would use other measures not directly related to the root of the noise.

Among several different ways of quantifying noise, following is a list of the one selected during the research experience at the ACRC.

Direct method:

1. Refrigerant pressure fluctuations.

   For this purpose, high speed pressure transducers measuring directly in the flow have been used.

Indirect methods:

2. Structural oscillations.

   For this purpose, accelerometers mounted on the surface of some components have been used.

3. Air-borne noise.

   Apparently, the easiest way of measuring is very complicate to actuate due to the background noise and other issues related to the position at which microphones can stand. This also makes the experiment difficult to duplicate.

To overcome some of these problems microphone measurements in anechoic chambers have been used.

4. EXPERIMENTAL FACILITY

4.1 EXPERIMENTAL APPARATUS

After a careful study of similar researches, an experimental apparatus suitable for this work has been realized.

The apparatus (figure 4.1) was created in order to quantify the noise coming from the expansion device based on different measurement techniques:

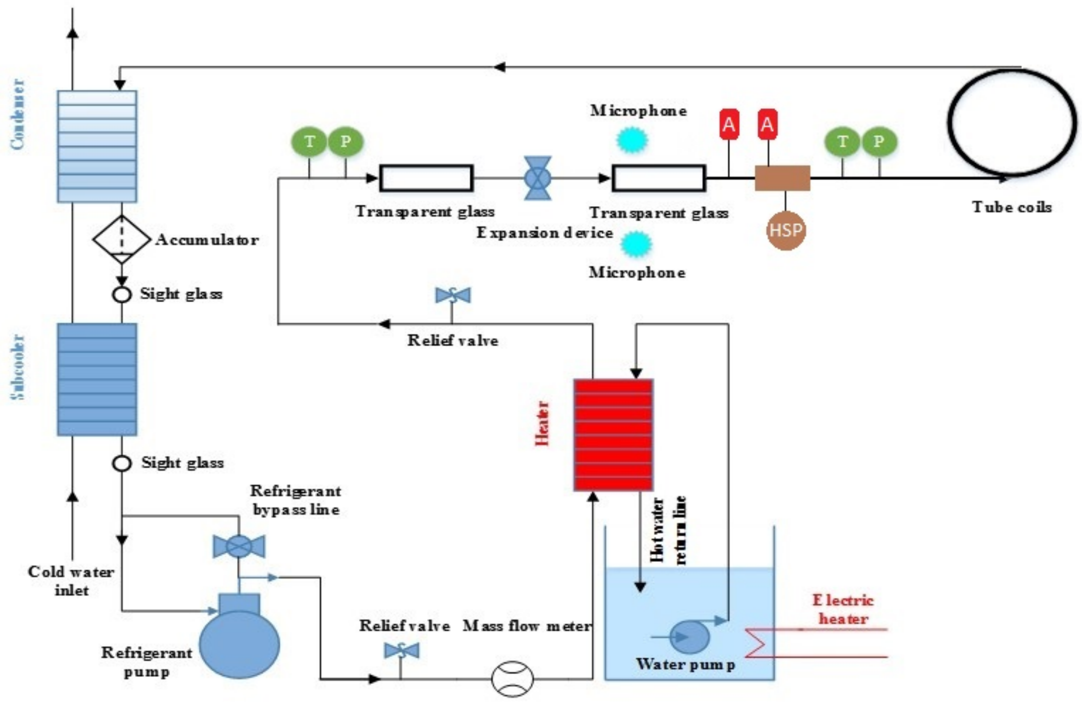

Figure 4.1: *Schematic of the Experimental Facility Designed for this Work.*

Indeed, it was created not only to measure the noise via air, but also to allow flow measurement of liquid (refrigerant) to characterize liquid flow through the expansion device. Through this experimental facility it is possible to measure temperatures, pressures, mass flow rate, accelerations, and noise near the expansion device using as a medium both air and liquid refrigerant.

In order to change more easily the conditions at the inlet and outlet of the expansion valve, the facility uses a pump instead of the usual compressor. This makes the change of all thermostatic variables a faster process to actuate.

A P-h diagram for this specific system is shown in figure 4.2.

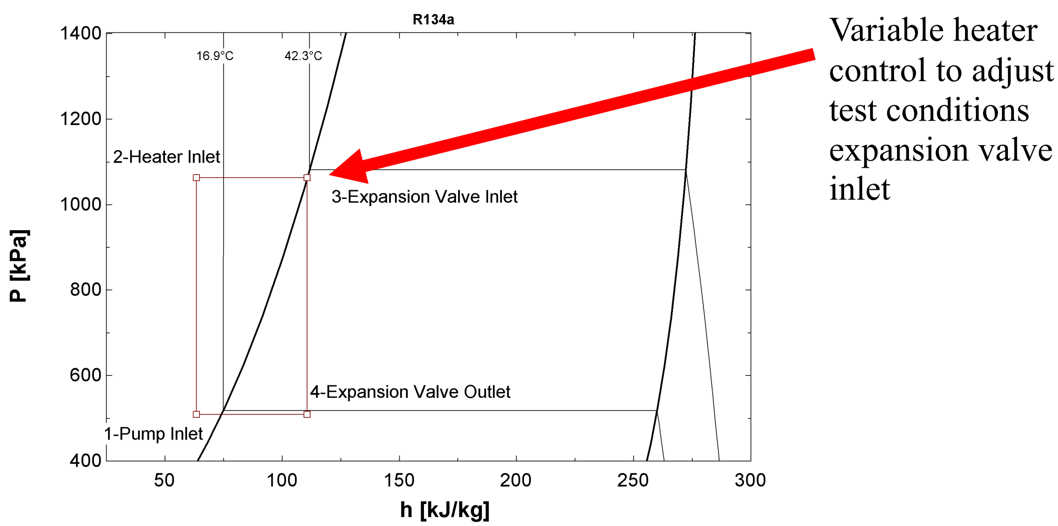

Figure 4.2: *P-h diagram of the Cycle for the System Used.*

Instead, always looking at figure 4.1, it is possible to see how the loop works. First sub-cooled refrigerant (R134a has been used for this work) goes into the pump (1), which dramatically increases the pressure of the working fluid. The following step (2) for the fluid is to enter the heater which increases the enthalpy at constant pressure. At this point the fluid goes through the expansion valve (3) - core of this project - where it is expanded. The expansion valve - a needle valve in this case - can be regulated in order to decrease the temperature of the refrigerant by how many degrees it is desired. At this stage (4), the fluid goes from a single phase (liquid) to two phases (liquid and gas), due to the sudden decrease in pressure. Then, the refrigerant goes again into the condenser and the sub-cooler to restart the whole cycle.

Now that the basic operation of the system is clear, a focus will be placed on some key features of the system.

First of all, safety. As said before the pump can increase the pressure of the refrigerant by several thousands kPa, that is why a safety relief valve has been placed soon after it. Moreover, this relief valve can be very important whenever the system - fully loaded - is placed in standby for a long time. Indeed, the increase in temperature can lead to an increase in the pressure inside the system which can be catastrophic for its integrity (considering the presence of plastic transparent glasses).

Another peculiar feature of this system is the tube coil after the expansion device, which is about 24 meters long with a diameter of 60 cm (>25x than the Inside Diameter of the pipe). The pipe type used for the whole system is about 0.5 in outside diameter and 0.43 in inside diameter (measures are given in inches to give better precision). The reason for this choice is to reduce sound-waves coming from the downstream components (see chapter 3). Indeed, the curvature and length of the pipe after the expansion valve allow for a bigger and smoother reduction of reflected sound waves. It is also important to consider that even if the tube is particularly long, there are no relevant pressure losses in the downstream of the expansion valve.

Figure 4.3: *View of the Experimental Apparatus.*

After the coil, there is the sub-cooler which is a heat exchanger fed by chilled water at 4-5 °C. A similar feature happens for the heater (2), it is fed by a water pump with warm water at about 40 °C. The water is then heated with an electric heater. The reason for which the electric heater is not in direct contact with the refrigerant is to prevent it from burning when passing through the heater.

Between steps (1) and (2) there is a mass flow meter that only measures single phase liquid flows, since the working fluid is sub-cooled before entering the pump (at around 8-9 °C), and this last one is obviously not changing the state of the refrigerant.

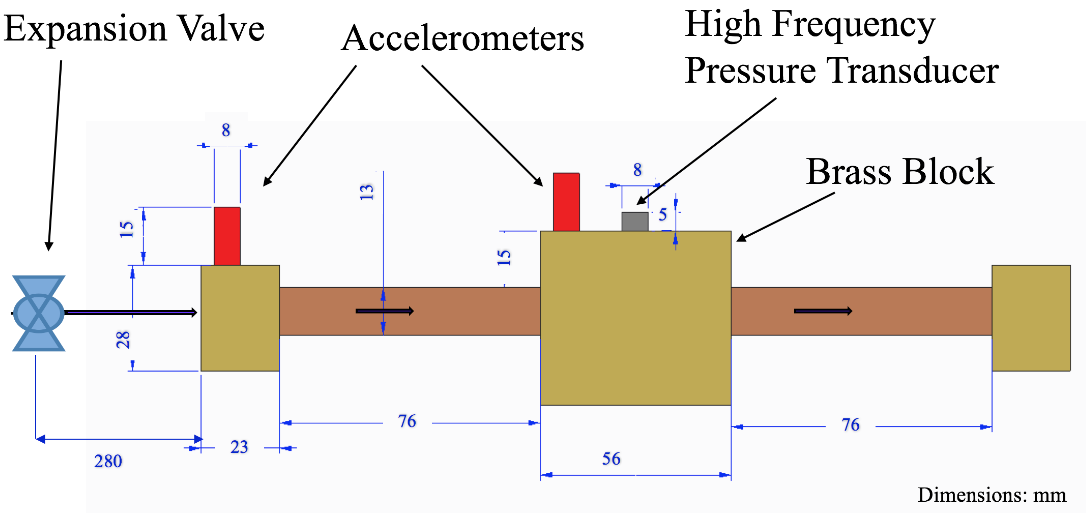

Figure 4.4: *Test Section Schematic.*

Now there will be a quick introduction to the test section (fig 4.4-4.7), in order for the reader to have a more general view of the system. The test section which is delimitated by states (3) and (4) is composed by two transparent sight glasses, before and after the expansion valve (a manually regulated needle valve), two temperature and two pressure transducers, two accelerometers, two microphones and one high frequency pressure transducer. The purpose of the sight glasses is to make visualization, in order to link specific noise patterns to flow patterns. The two temperature and pressure transducers are used to control and regulate the system conditions.

A brass block was also used to reduce structure vibrations where the high frequency pressure transducer was mounted.

Regarding the microphones, the accelerometers and the high frequency pressure transducer, more will be stated in paragraph 4.3.

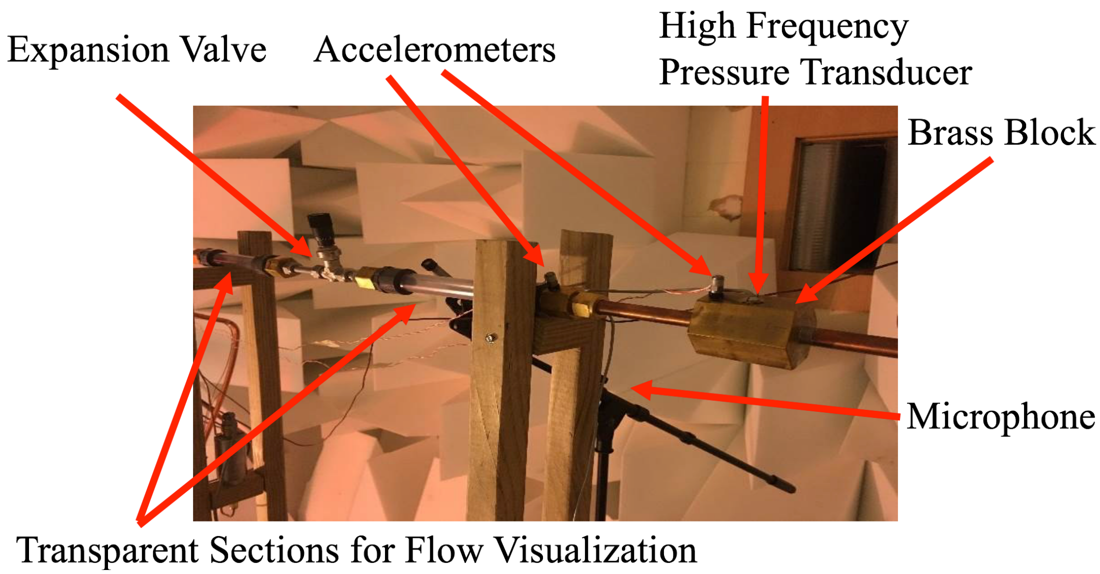

Figure 4.5: *View of the Real System Test Section.*

Figure 4.6: *Needle Valve.*

A fresh look is now given to the regulation of all working parameters. This system was built in such a way that mass flow rate, pressures and temperatures could be changed as preferred (see fig. 4.2). First, the sub-cooler. It can be regulated by closing the valve fed by cold water, this operation however, can create some issues with the pump and mass flow rate, indeed, if the refrigerant is not sub-cooled it would be in two phases and it would damage the pump for cavitation. Furthermore, the mass flow rate would not be able to work, due to the presence of gas. In order to stay in the sub-cooled region, it is possible to look at chart 4.1.

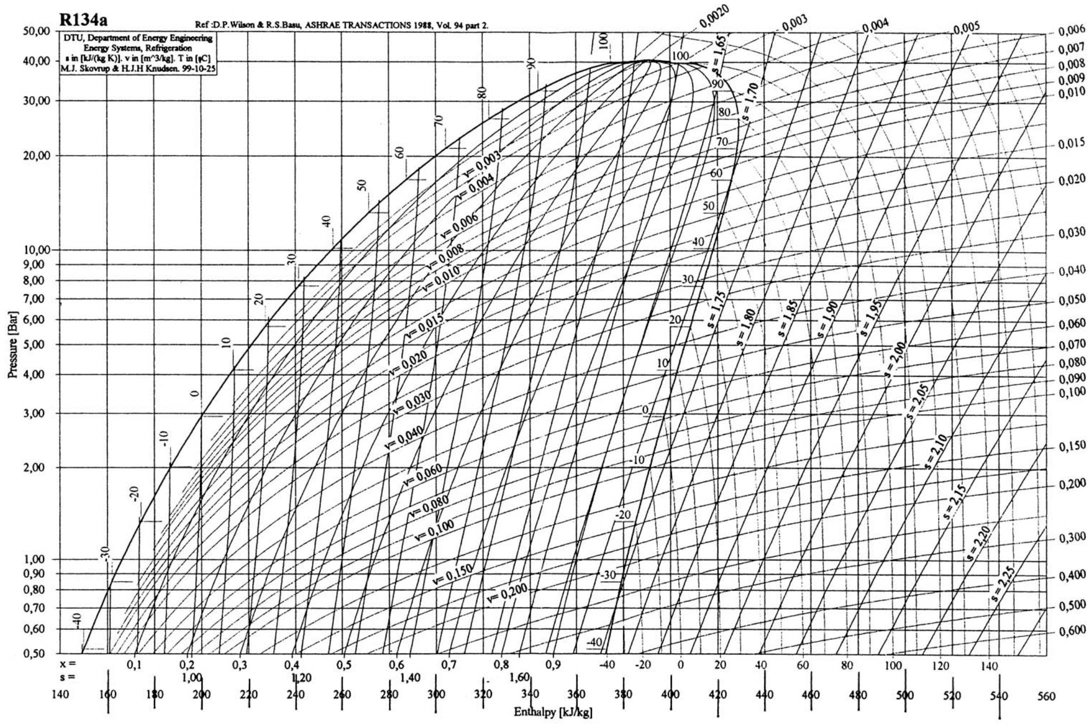

***Chart 4.1**: R134a P-h Diagram.*

Another regulation can be made by adjusting the speed of the pump, which is given by an increase in frequency of rotation, and hence in an increase of the mass flow rate and inlet pressure at the expansion valve.

As with the previous heat exchanger, it is possible to regulate the temperature of the heater, where an increase in temperature could lead to a change in enthalpy and hence in the state of the refrigerant. Furthermore, attention must be paid to the increase in temperature to avoid a change of phases.

Finally, the most important regulation is given by the expansion valve. As stated in chapter 2, based on the opening of the expansion valve it is possible to increase or decrease the pressure drop and hence the temperature. In particular, closing the expansion valve leads to a bigger decrease in pressure and thus, temperature; while the opposite is true when the valve is opened more.

4.2 ANECHOIC CHAMBER

A long time has been spent in chapter 3 to describe noise and how to reduce or account for the waves' reflection in the measurements. This section of the thesis, instead, will be used to show a major room of the experimental facility: the anechoic chamber (figure 4.7).

The term anechoic comes from Greek, where 'an-echoic' means without echo. Indeed, the anechoic chamber is a room where echo and reflective waves are not transmitted. It is a room designed to completely absorb reflections of sound. Moreover, there is a strong insulation from the outside because other than absorbing reflection waves, it can also block external sound waves from entering the room.

For these reasons an anechoic chamber has been used for this work. We were able to isolate the room and have a sound pressure level of about 30 dB. Which is much lower than the background noise present in other rooms (around 60 dB). Indeed, to better isolate the expansion valve noise, all the motors components have been placed in another room (figure 4.3).

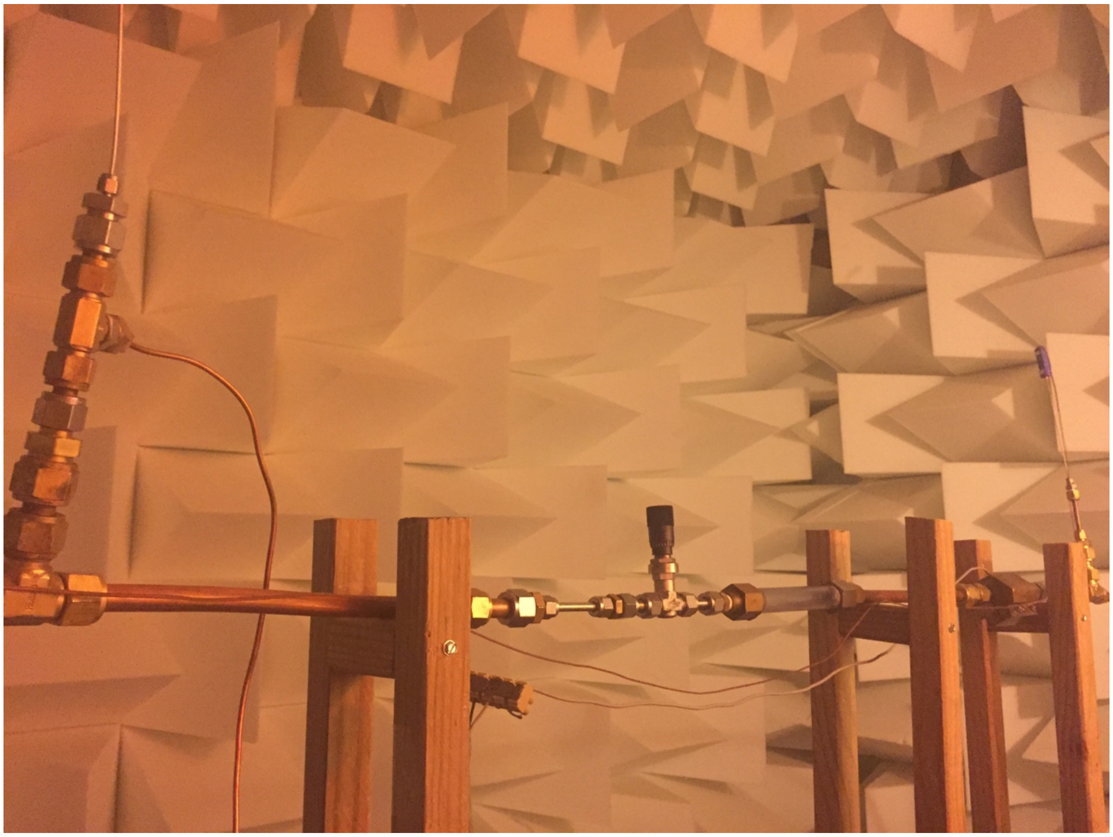

Figure 4.7: *Anechoic Chamber.*

Some features of the anechoic chamber:

- It has been created by using foam cones glued to every wall;
- On the floors, the cones are removable (semi-anechoic chamber), for whenever major changes must be made to the test section. However, before conducting any experiment, the cones are placed on the floor so that it becomes again a full anechoic chamber;
- Finally, it is important to say that a door is present to allow the entrance of workers. This door is usually closed when experiments are running.

In order to isolate vibrations and reduce even more the absorption of sound waves, a wood structure was used to support all the pipes instead of the classical metal frame. By doing so and by removing every kind of metal present in the anechoic chamber sounds wave reflections and any interference have been minimalized.

4.3 MEASURMENT INSTRUMENTATION

The present paragraph is targeted at describing the main measurement instruments used to build the system. The characteristic instruments used are: an ICP power supply, two single-axis accelerometers, a high frequency pressure transducer, two pre-polarized microphones and preamplifiers and a high frequency data acquisition. The reader should refer to figure 4.4, while reading the next paragraphs, to better understand the use of each instrument.

4.3.1 SINGLE-AXIS ACCELEROMETERS & ICP POWER SUPPLY

Going back to Paragraph 3.2, one of the indirect measurement methods for noise is to investigate structural oscillations. This is done by using two single-axis accelerometers (figure 4.8) mounted on the system structure.

Figure 4.8: *Accelerometer Single Axis PCB Model 352C67*.

Both are placed after the expansion valve using a special wax for accelerometers. The first one is mounted right after the sight glass, where the vibrations are higher, while the other is mounted on the brass block, close to the high frequency pressure transducer. The are two main reasons leading to these decisions: to understand the impact of vibrations on pressure fluctuations ,and being able to link pressure oscillations with vibrations. This last purpose could only be accomplished by mounting both the accelerometer and the high frequency pressure transducer on the brass block.

Finally, both accelerometers are connected to a dedicated ICP power supply (figure 4.9). The power supply itself is used also as a data reader and is thus connected to a high frequency data acquisition instrument that will be described in paragraph 4.3.4.

Figure 4.9: *Power Supply PCB Model 482A.*

Manuals for these instruments can be found easily on the website of the manufacturer. Here only a few details will be given for the single-axis accelerometers:

- Sensitivity: 100 mV/g;
- Measurement range: 50 g peak;
- Frequency range: 0.5-10 kHz.

4.3.2 PIEZORESISTIVE HIGH FREQUENCY PRESSURE TRANSDUCER

To directly measure pressure fluctuations of the refrigerant, a high frequency pressure transducer was used (figure 4.10).

This piece of equipment is screwed to the brass block and its measuring surface is in direct contact with the working fluid.

The pressure transducer is then connected to a differential voltage amplifier (figure 4.11), which as the name suggests amplifies the signal up to 1000 times, going from mV to Volts.

Figure 4.10: *High frequency pressure transducer Endevco Model 8510B-200.*

Some important features of this pressure transducer are:

- Pressure range: 0-200 psig;
- Sensitivity: 1.5 mV/psi;
- Resonance frequency: 320 kHz.

The reason for using a high frequency pressure transducer instead of a more conventional one is that it can be utilized to measure dynamically. It is possible to measure fluctuations in the range of sound waves up to 20 kHz.

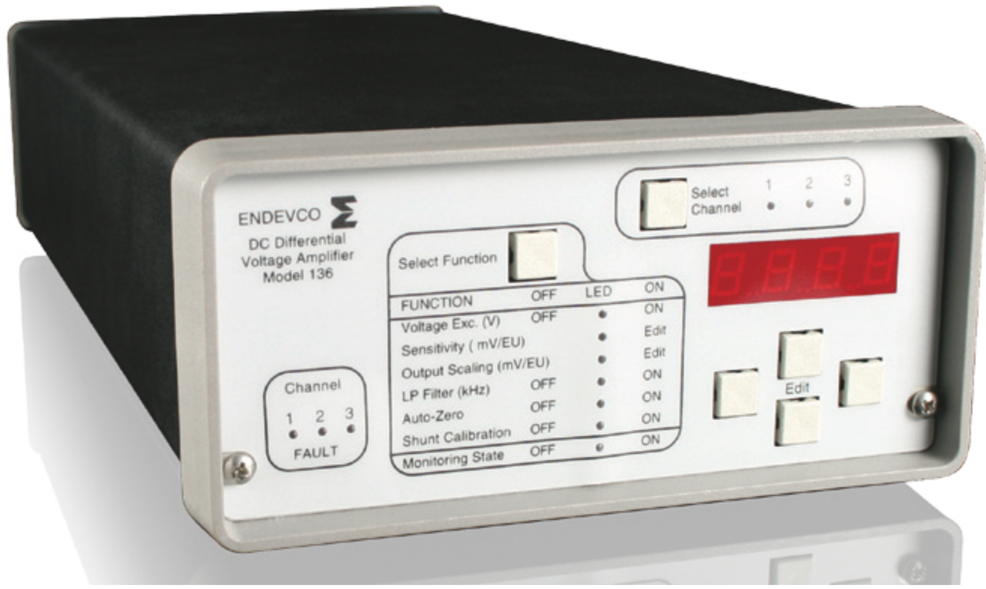

Figure 4.11: *Differential Voltage Amplifier Endevco Model 136.*

Finally, some technical specifications for the differential voltage amplifier for future reference:

- Three-channel DC differential voltage amplifier;
- 200 kHz bandwidth (-3dB corner);
- Auto-zero and shunt calibration;
- Gain range 0 to 1000;
- Four selectable excitation voltage levels;
- RS-232 serial interface;
- 12 VDC power option;
- Built-in 4-pole Butterworth low pass filter;
- Optional low-pass filter module with different corner frequencies.

4.3.3 PREPOLARIZED MICROPHONES

The two prepolarized microphones used for this research are composed by a preamplifier excited with an IEPE input and a microphone head. They have a sampling frequency going from 20 Hz to 12 kHz, sufficient for the purposes of this work (see Chapter 5). Moreover, measurements done with these instruments are accurate enough with a sensitivity of 31.6 mV/Pa.

Figure 4.12*: Preamplifier and Microphones, Brüel & Kjær Type 2671 and 4188-A-02.*

4.3.4 HIGH-FREQUENCY DATA ACQUISITION DEVICE

All the above transducers are connected to the high frequency data acquisition device, NI 9234 (figure 4.13). It is a very advanced instrument capable of handle big amounts of data at high frequencies. Following are some its features:

- Maximum sampling rate: 51.2 kS/s per channel;
- Number of channels: 4;

- Software selectable IEPE signal conditioning (0 or 2 mA).

Figure 4.13: *National instruments NI9234.*

The data signal acquired by this device, will then be pre-analyzed and saved using a LabVIEW program created specifically for purpose.

5. ANALYSIS METHODS

After the acquisition of data from the LabVIEW program, comes the need to analyze them.

For these kind of data the goal is to discretize the different signals into multiple sinusoidal components, going from time domain to frequency domain. This process is done by means of the Discrete Fourier Transform (DFT).

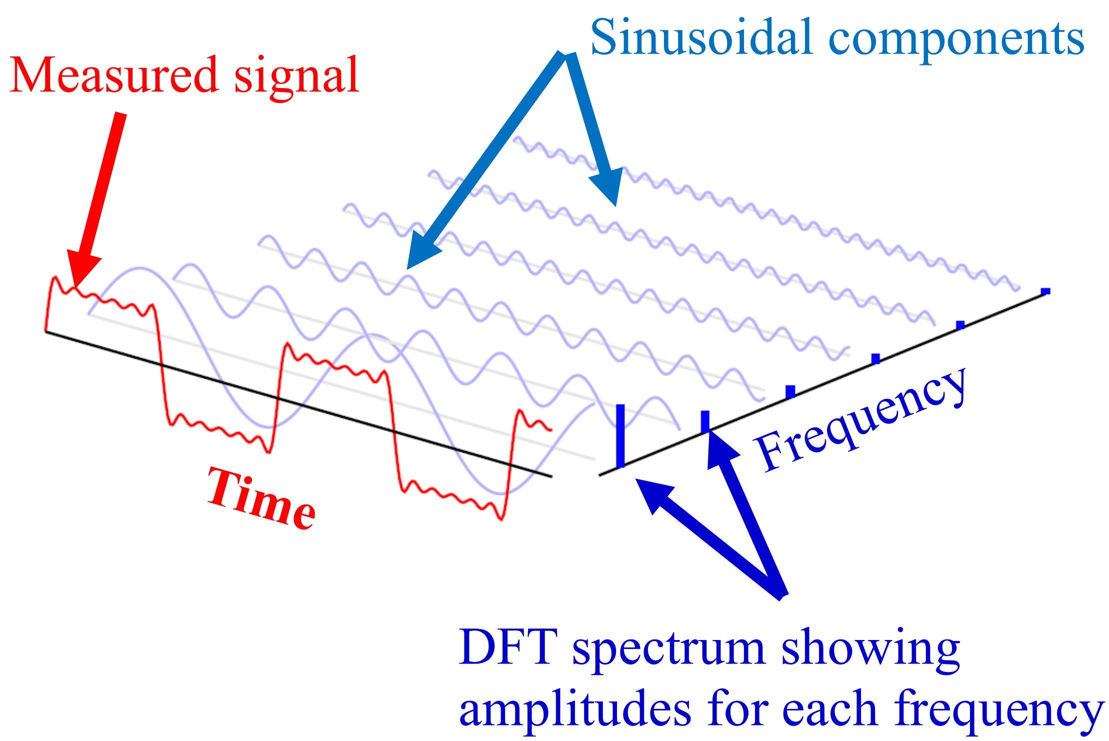

Figure 5.1: *Discretization of a Time Signal into Multiple Sinusoidal Components.*

Def. (DFT) [19]: given a sequence $x = [x_0, \dots, x_{n-1}]^T$, its discrete Fourier transform, is the sequence $y = [y_0, \dots, y_{n-1}]^T$ given by:

$$y_m = \sum_{k=0}^{n-1} x_k w_n^{mk}, \qquad m = 0,1,\dots,n-1.$$

One way to compute the DFT is to use the Fast Fourier Transform (FFT) which is an optimized algorithm that takes advantage of some redundancies and symmetries present in the original DFT algorithm.

To do so, a Matlab code was written and utilized. This code is able to discretize the pressure and acceleration signals from time to frequency domains. For the purposes of this work the range of interest is 2-6 kHz.

Following is the script of the code used:

```
clear X Y Z mfl tmpin tmpout pin pout

X=beforebrass1;
Y=brassblock1;
Z=pressure1;
mfl=mflow1;
tmpin=tempin1;
tmpout=tempout1;
pin=pin1;
pout=pout1;


Fs = 12830;             % Sampling frequency
T = 1/Fs;               % Sampling period
L = 384900;             % Length of signal
t = (0:L-1)*T;
figure
plot(t,X,t,Y)
title('Acceleration')
xlabel('t(seconds)')
ylabel('Acceleration(m/s^2)')
legend('Before Brass Block','Brass Block')
```

```
XX = fft(X);
YY = fft(Y);
P2 = abs(XX/L);
P3 = abs(YY/L);
P1 = P2(1:L/2+1);
P0 = P3(1:L/2+1);
P1(2:end-1) = 2*P1(2:end-1);
P0(2:end-1) = 2*P0(2:end-1);
f = Fs*(0:(L/2))/L;
figure
plot(f(4:end),P1(4:end),f(4:end),P0(4:end))
title('Single-Sided Amplitude Spectrum of the
Acceleration')
xlabel('f (Hz)')
ylabel('Amplitude (m/s^2)')
legend('Before Brass Block','Brass Block')

Fs = 12830;              % Sampling frequency
T = 1/Fs;               % Sampling period
L = 384900;               % Length of signal
t = (0:L-1)*T;
figure
plot(t,Z)
title('Pressure')
xlabel('t(seconds)')
ylabel('Pressure(kPa)')
ZZ = fft(Z);
P5 = abs(ZZ/L);
P4 = P5(1:L/2+1);
P4(2:end-1) = 2*P4(2:end-1);
f = Fs*(0:(L/2))/L;
figure
plot(f(4:end),P4(4:end))
title('Single-Sided Amplitude Spectrum of the Pressure')
xlabel('f (Hz)')
ylabel('Amplitude (kPa)')


time=mfl(:,1);
mf=mfl(:,2);
figure
plot(time,mf)
ylim([5 25])
title('Mass Flow')
xlabel('t (s)')
ylabel('Mass flow rate (g/s)')
```

```
tmpin=tempin1;
tmpout=tempout1;
time=tmpin(:,1);
tmin=tmpin(:,2);
tmout=tmpout(:,2);
figure
plot(time,tmin,time,tmout)
ylim([5 60])
title('Temperature')
xlabel('t (s)')
ylabel('Temperature (C)')
legend('Temperature inlet','Temperature outlet')

pin=pin1;
pout=pout1;
time=pin(:,1);
prin=pin(:,2);
prout=pout(:,2);
figure
plot(time,prin,time,prout)
ylim([300 1300])
title('Pressure')
xlabel('t (s)')
ylabel('Pressure (kPa)')
legend('Pressure inlet','Pressure outlet')
```

Examples of FFT analysis for acceleration and pressure measurements are reported below respectively:

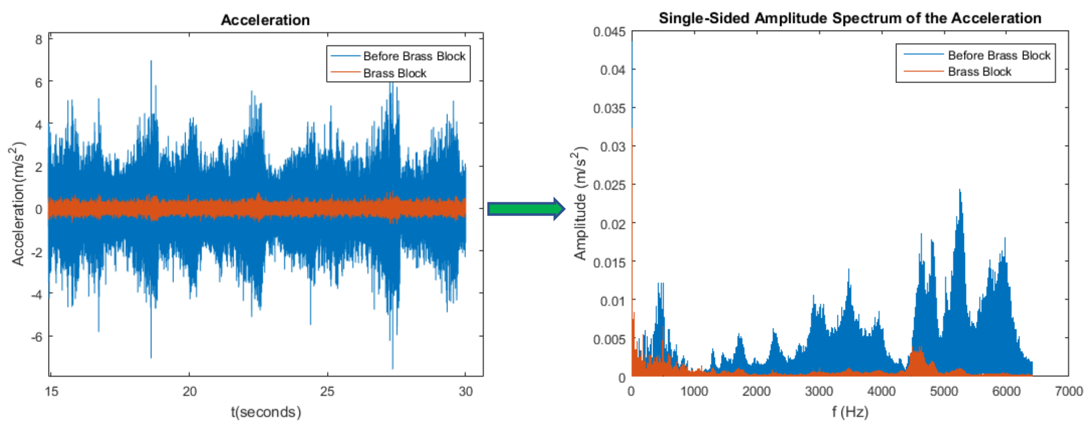

Figure 5.2: *Example of FFT Analysis for Acceleration Measurements.*

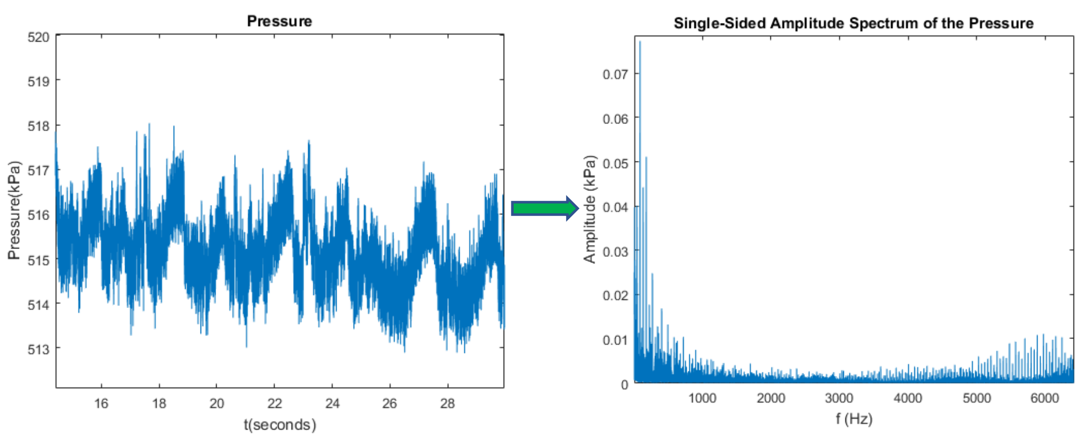

Figure 5.3: *Example of FFT Analysis for Pressure Measurements.*

6. EXPERIMENTAL RESULTS & CONCLUSIONS

6.1 EXPERIMENTAL RESULTS

Experiments have been conducted under different working conditions, however, since this is not the main objective of the thesis, only an example will be reported.

In this chapter the comparison between two working conditions for the system will be shown. One condition called 'high' mass flow rate and the other, 'low' mass flow rate. These settings (table 6.1) are obtained by changing the speed of the pump and by regulating the expansion valve in order to maintain all the other variables constants (or close one to the other).

	Setting 1 (high mass flow rate):	Setting 2 (low mass flow rate):
Pump Frequency	**10 Hz**	**7 Hz**
Mass Flow Rate	**14.6 g/s**	**11.6 g/s**
T inlet Pump	**8.4 °C**	**8.5 °C**
T inlet Expansion Valve	**42.3 °C**	**42.3 °C**
T outlet Expansion Valve	**17.4 °C**	**16.9 °C**
P inlet Expansion Valve	**1071 kPa**	**1063 kPa**
P outlet Expansion Valve	**517 kPa**	**509 kPa**

Table 6.1: *Example of Two Working Settings.*

The purpose of this experiment is to compare the acceleration (figure 6.2) and pressure (figure 6.3) patterns for these two settings.

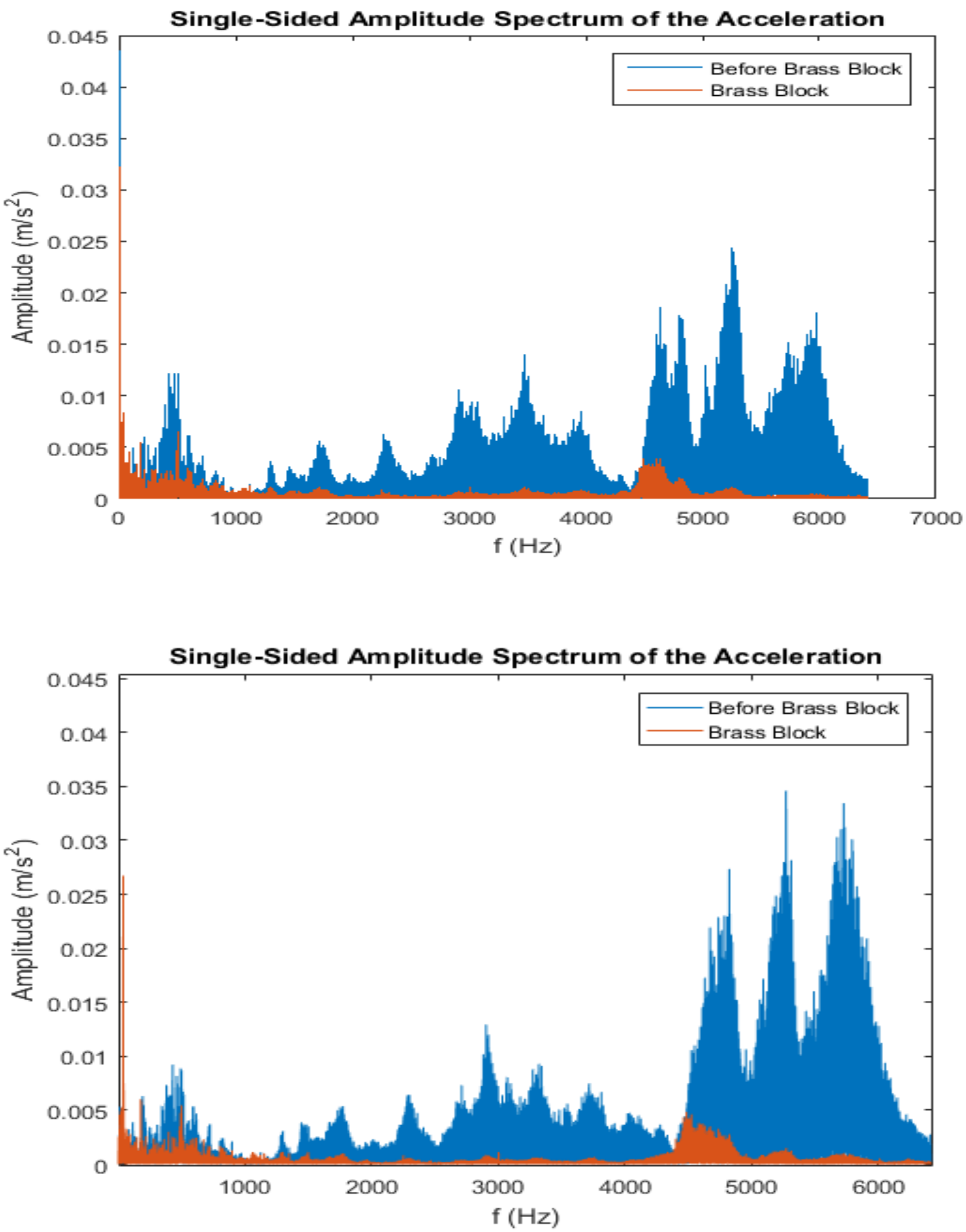

Figure 6.2*: FFT of Acceleration for Setting 1 (Up) and 2 (Down).*

Looking at both figures, it is possible to see that there are similar frequency distributions patterns for both settings and unexpected higher vibrations in the range of 4.5-6 kHz for setting 2.

Instead, looking at figure 6.3 it is clear that while the frequency distribution patterns are again similar for both settings, there are lower pressure fluctuations in the range of 5-6 kHz and more intensified pressure fluctuations for frequencies lower than 400 Hz for setting 2.

Trying to compare the acceleration and pressure signals for setting 1, it is evident that both graphs show high amplitude contributions at <1 kHz and >4.5 kHz. Moreover, it is important to notice that acceleration measurements done on the brass block resemble the pressure signal more closely than measurements done before the brass block.

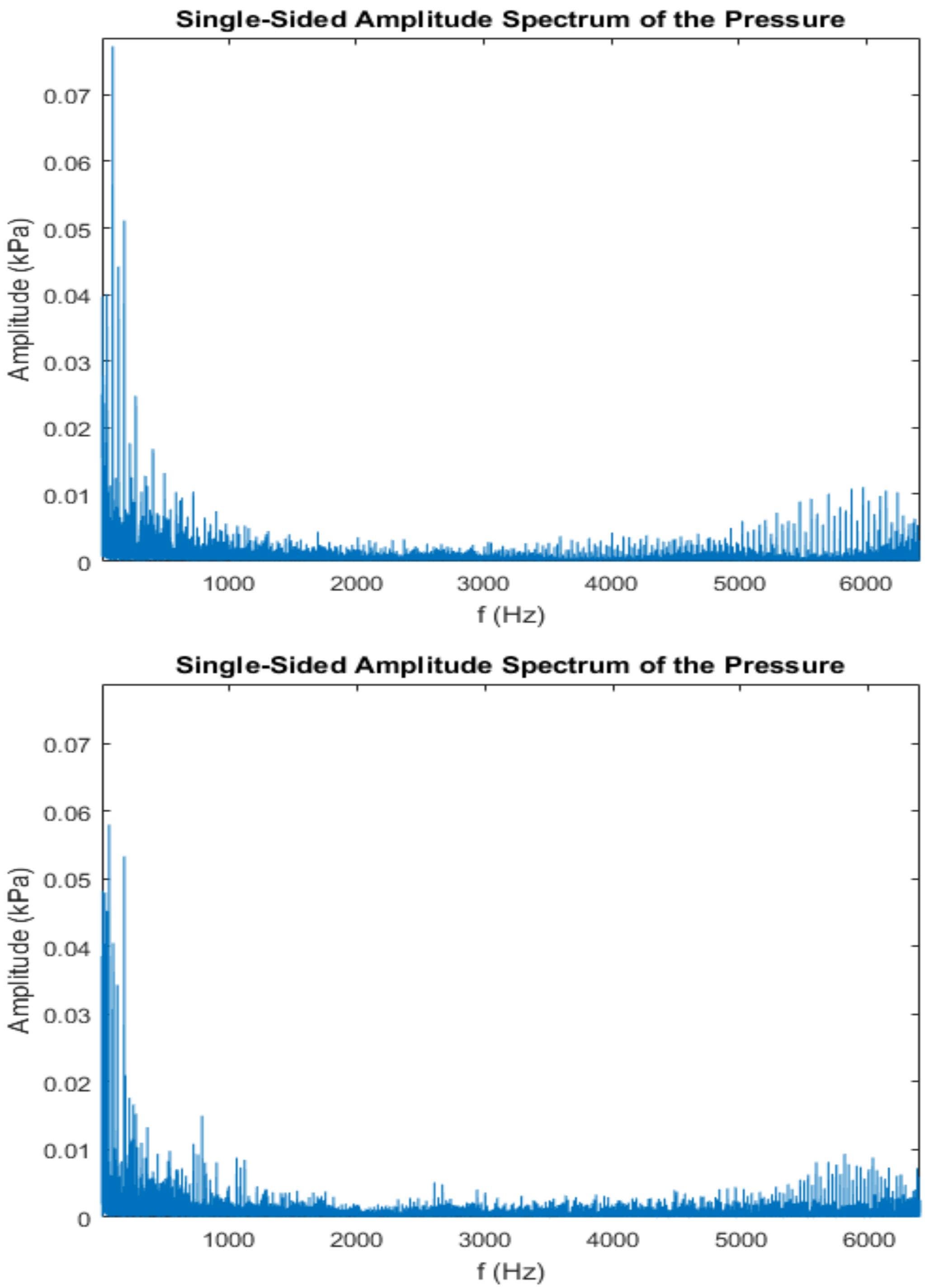

Figure 6.3: *FFT of Pressure for Setting 1 (Up) and 2 (Down).*

Some flow visualization (figure 6.4) has also been done using a high speed camera. It shows how the flow regime changes due to two-phase feeding of the expansion valve (happening in this specific case).

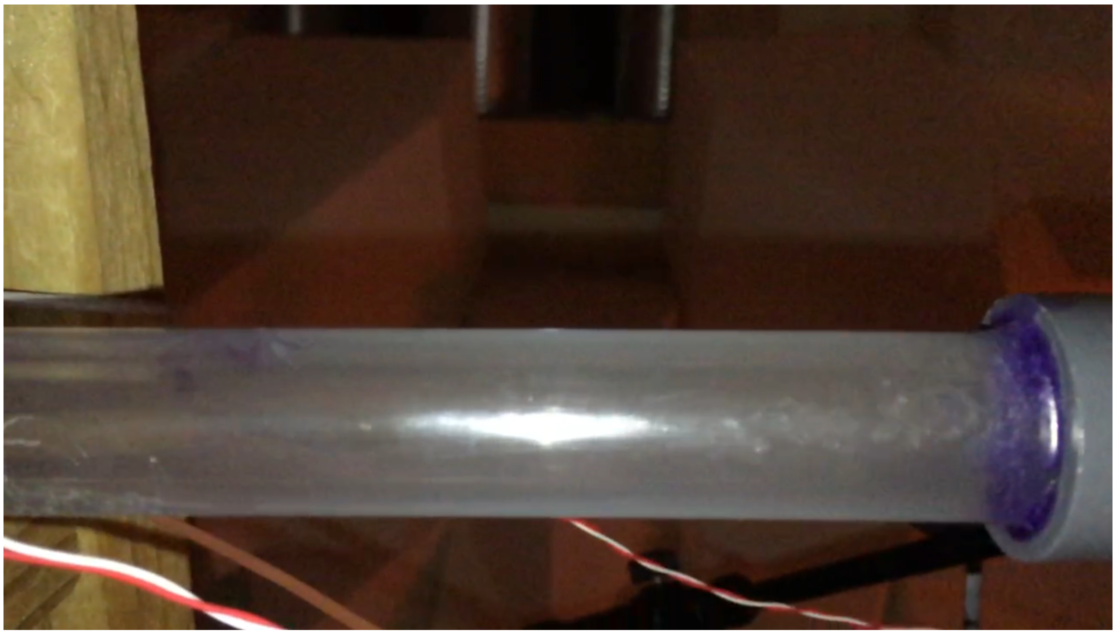

Figure 6.4: *Flow Visualization.*

6.2 CONCLUSIONS AND RECOMMENDATIONS

This research lead to various findings regarding ways in which noise can be measured, and how facility settings can affect the results. Thus, here are some final suggestions for setting up the experiment:

- Anechoic chamber: it was found that to make the noise absorption more efficient it is opportune to remove every structure with sharp angles and flat surfaces. Moreover, it is better to avoid metallic frames and structures due to high vibrations measured that would result in noise creation and propagation. For this reason, a thin wood supportive frame was chosen so it helps to absorb noise. In addition, it is advantageous to avoid any walking paths above the floor's foam cones (which due to vibrations may alter the experimental results) and instead it can be useful to remove the foam cones on the floor whenever it is required to enter the chamber to adjust the settings and replace the cones before taking measurements. In this way, a fully anechoic chamber capable of absorbing noise more efficiently was created.

- Piping: it was ascertained that every curvature of the pipes - especially sharp angles bigger than 45 degrees - resulted in torsional moment applied by the working fluid on the pipes (like what happens with a garden sprinkler). In this case noise is created not only by the disturbed flow but also by the vibrations created (structure born noise). To avoid these kinds of problems, whenever a change of direction was needed, a smooth, gradual curve was used instead of sharp angles.

  Furthermore, it is useful to apply high inertia to the pipes to avoid unwanted vibrations caused by the system's structure, thus, isolating the noise produced by the expansion valve.
- Finally, thanks to the collaboration with project sponsors it was understood that the use of accelerometer can be a delicate theme. There are too many variables that make it almost impossible to convert acceleration into noise. For this reason, it has been decided to avoid the use of accelerometers

and instead use high frequency pressure transducers to relate the pressure fluctuations with the noise measured by the microphones. Regarding this last piece of equipment, microphones are placed at distances that best mimic the perception of noise by the user.

- Pump: regarding the pump, it was discovered that when it operates with a frequency below 20Hz, it is not possible to maintain a perfect steady state condition (except for particular cases, which were analyzed in the thesis). For this reason, it is suggested to use a frequency higher than 20Hz such that it is possible to stay in the range of sound frequencies, and obtain a more realistic steady state condition.

In conclusion, the objective of this thesis was to provide a basis for future scientists who would like to come against the problem of noise produced from expansion devices. First the state of the art, mentioned previous research findings related to this thesis. Second, basic theories regarding noise and expansion devices were defined and explained. Subsequently the thesis demonstrated how the laboratory was designed and setup (the refrigerant used was R134a), including pressure, acceleration, and microphone measurements. In particular, the importance of the anechoic chamber has been shown. After it has been revealed the usefulness of the FFT to convert time domain measurements to frequency domain signals for detailed analysis. Initial measurements have been conducted and similarities between pressure and acceleration measurements were investigated in frequency range of interest (<

6 kHz). Lastly, preliminary flow visualization results are available to link flow regimes to noise measurements.

For future work, it is suggested to:

- Conduct microphone measurements and compare them to pressure signals;
- Investigate different valve settings and operating conditions;
- Conduct experiments with different refrigerants (e.g. R410A, R32, R1234yf) to study effects of thermophysical properties;
- Investigate additional expansion device types (orifice geometries) and the effects of adjacent components;
- Visualize flow and relate observed flow regimes to recorded sound emissions/vibrations;
- Develop noise mitigation strategies, e.g. absorbing materials, geometry optimization, etc.

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